TRANSACTIONS

OF

THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

1919

VOL. XXVII 1919

TRANSACTIONS

OF

THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

EDITED BY THE SECRETARY

PRINCIPAL OFFICE OF THE SOCIETY
ENGINEERING SOCIETIES BUILDING
29 WEST 39TH STREET, NEW YORK, N. Y., U. S. A.

ROBERT GIBSON
DIRECTOR OF PRINTING
WASHINGTON, D. C.

NOTICE

The Society as a body is not responsible for any statements of fact or opinion contained in the following papers and discussions, such responsibility resting entirely with those making the statements.

CONTENTS

INDEX TO PAPERS AND DISCUSSIONS.

Thursday, November 13, 1919.

Friday, November 14, 1919.

OFFICERS OF THE SOCIETY

President.

WASHINGTON L. CAPPS.

Past Presidents and Life Members.

FRANCIS T. BOWLES, STEVENSON TAYLOR.

Past President and Life Associate Member.

ROBERT M. THOMPSON.

Honorary Vice-Presidents.

F. L. FERNALD, D. W. TAYLOR,
F. E. KIRBY, W. M. McFARLAND.

Vice-Presidents.

Term expires December 31, 1920:

W. F. DURAND, W. J. BAXTER,
H. L. FERGUSON, H. A. MAGOUN.

Term expires December 31, 1921:

LEWIS NIXON, C. H. PEABODY,
H. I. CONE, J. W. POWELL.

Term expires December 31, 1922:

A. P. NIBLACK, H. D. GOULDER,
R. M. WATT, C. P. WETHERBEE.

Members of Council.

Term expires December 31, 1920:

H. C. SADLER, D. H. COX, A. C. PESSANO,
F. L. DuBOSQUE, W. A. DOBSON, C. F. BAILEY.

Term expires December 31, 1921:

W. D. FORBES, R. H. M. ROBINSON, J. H. MULL,
ANDREW FLETCHER, W. G. COXE, ROBERT HAIG.

Term expires December 31, 1922:

J. H. LINNARD, W. L. R. EMMET, W. J. DAVIDSON,
C. A. McALLISTER, J. H. GARDNER, H. P. FREAR.

Associate Members of Council.

Term expires December 31, 1920:

H. L. ALDRICH, H. H. RAYMOND.

Term expires December 31, 1921:

E. M. BULL, C. M. WALES.

Term expires December 31, 1922:

R. A. C. SMITH, A. W. GOODRICH.

Executive Committee.

WASHINGTON L. CAPPS, *Ex-Officio.*

STEVENSON TAYLOR, W. M. McFARLAND, J. W. POWELL,
ANDREW FLETCHER, F. L. DuBOSQUE, D. H. COX, *Ex-Officio.*

Secretary-Treasurer.

DANIEL H. COX.

ARTICLES OF INCORPORATION

THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

State of New York, }
City and County of New York. }

We, the subscribers, Wm. H. Webb, Chas. H. Cramp, George E. Weed, H. Taylor Gause, William T. Sampson, Horace See, Frank L. Fernald, Francis T. Bowles, Washington L. Capps, Edwin D. Morgan, George W. Quintard, Harrington Putnam and Jacob W. Miller, being persons of full age and citizens of the United States, of whom a majority—namely, William H. Webb, George E. Weed, Horace See, Edwin D. Morgan, George W. Quintard, Harrington Putnam, Frank L. Fernald, and Jacob W. Miller are citizens of and residents of and within this State, desiring to associate ourselves for scientific purposes under, and pursuant to, an Act of the State of New York providing for the incorporation of benevolent, charitable, scientific, and missionary societies, passed April 12, 1848, and the several acts amending or supplementing the same, do hereby, in accordance with the requirements thereof, certify as follows:—

First. The name of title by which the Society shall be known in law is The Society of Naval Architects and Marine Engineers.

Second. The particular business and objects of such Society are the promotion of practical and scientific knowledge in the arts of shipbuilding and marine engineering and the allied professions, and in furtherance of this object, to hold meetings for social intercourse among its members, and the reading and discussion of professional papers, and to circulate by means of publication the knowledge thus obtained.

Third. The number of directors, trustees, or managers to manage the Society shall be seven, and shall consist of a President, a Secretary, and five Members of Council.

Fourth. The names of the trustees, directors, or managers of the Society for the first year of its existence are:—President, Clement A. Griscom; Secretary, Washington L. Capps; Members of Council, Francis T. Bowles, H. Taylor Gause, Chas. H. Loring, Lewis Nixon, Harrington Putnam.

Fifth. The business of the Society is to be conducted, and its place of business and principal office is to be located, in the City and County of New York.

In Witness Whereof we have made, signed, and acknowledged this Certificate, this 28th day of April, 1893.

WILLIAM H. WEBB.
CHAS. H. CRAMP.
H. T. GAUSE.
GEORGE E. WEED.
W. T. SAMPSON.
HORACE SEE.
F. L. FERNALD.

FRANCIS T. BOWLES.
W. L. CAPPS.
E. D. MORGAN.
GEORGE W. QUINTARD.
HARRINGTON PUTNAM.
J. W. MILLER.

City and County of New York, *ss:*

On this 28th day of April, 1893, before me personally appeared William. H. Webb, Charles H. Cramp, H. Taylor Gause, George E. Weed, William T. Sampson, Horace See, Frank L. Fernald, Francis T. Bowles, Washington L. Capps, and Edwin D. Morgan, to me known and known to me to be the persons described in and who executed the foregoing certificate, and severally acknowledged to me that they executed the same.

James Forrester,
Notary Public, Kings Co., Cert. N. Y. Co.

City and County of New York, *ss:*

On this 1st day of May, 1893, before me personally appeared George W. Quintard and Harrington Putnam, to me known and known to me to be the individuals described in and who executed the foregoing certificate, and they severally acknowledged to me that they executed the same.

James Forrester,
Notary Public, Kings Co., Cert. N. Y. Co.

City and County of New York, *ss:*

On this 9th day of May, 1893, before me personally appeared Jacob W. Miller, to me known and known to me to be one of the individuals described in and who executed the foregoing certificate, and he duly acknowledged to me that he executed the same.

James Forrester,
Notary Public, Kings Co., Cert. N. Y. Co.

(Endorsed.)

Upon reading the within Certificate for the Incorporation of the Society of Naval Architects and Marine Engineers, I hereby approve and consent to the incorporation thereof and the within Certificate and filing thereof, and direct that the same be filed in the office of the Clerk of the City and County of New York.

Dated New York, May 10, 1893.

Edwd. Patterson,
Justice of the Supreme Court in the State of New York
in and for the City and County of New York.

INTRODUCTORY PROCEEDINGS

Minutes of the Twenty-Seventh Annual Meeting of the Society of Naval Architects and Marine Engineers, Held at the Engineering Societies Building, New York City, Thursday and Friday, November 13 and 14, 1919.

FIRST SESSION

Thursday Morning, November 13, 1919

The President, Rear Admiral Washington L. Capps, called the meeting to order at 10.20 a.m.

The President:—Gentlemen, the thirty-seventh meeting of the Society of Naval Architects and Marine Engineers will please come to order. As usual, we will begin our proceedings by the presentation of the report of the Secretary-Treasurer, Daniel H. Cox.

REPORT OF SECRETARY-TREASURER.

November 7, 1919.

To the Council of The Society of Naval Architects and Marine Engineers:—

Gentlemen:—I have the honor to submit the following report showing the condition of the Society at the close of the fiscal year ended October 31, 1919.

The membership of the Society as at October 31, 1919, was as follows:—

Class of Members	Membership November 1, 1918.	Elected at 1918 Meeting.	Promotion, Associate to Member.	Promotion, Junior to Member.	Promotion, Junior to Associate.	Membership at close Annual Meeting, 1918.	Promotion, Member to Life Member.	Promotion, Associate to Life Associate.	Promotion, Junior to Member.	Reinstated, 1918-1919.	Failed to Qualify.	Deaths, 1918-1919.	Resignations, 1918-1919.	Suspended, 1918-1919.	Error 1918 Report.	Membership, October 31, 1919.
Members........	709	308	+8	+3	—	1028	—12	—	+1	2	1	7	13	9	—1	986
Associates.......	202	136	—8	—	+5	335	—	—1	—	—	—	3	11	4	+1	315
Juniors..........	37	33	—	—3	—5	62	—	—	—1	—	—	1	—	—	—	60
Life Members....	4	—	—	—	—	4	+12	—	—	—	—	—	—	—	—	16
Life Associates...	6	—	—	—	—	6	—	+1	—	—	—	—	—	—	—	7
Honorary Members	1	—	—	—	—	1	—	—	—	—	—	—	—	—	—	1
" Associates	1	—	—	—	—	1	—	—	—	—	—	—	—	—	—	1
Totals.......	960	477	—	—	—	1437	—	—	—	2	1	11	24	13	—	1386

FINANCIAL

RECEIPTS.

From November 1, 1918, to October 31, 1919, inclusive.

Entrance fees of new members for the year ending October 31, 1919		$4,095.00
Dues for the year ended December 31:—		
1919	$11,575.00	
1918	815.00	
1917	135.00	
1916	75.00	
1915	50.00	
		12,650.00
Dues paid in advance:		
1920	$103.00	
1921	10.00	
		113.00
Entrance fees and dues paid in advance by proposed members		57.00
Dues in arrears—reinstated members		110.00
Life membership entrance fee		160.00
Total receipts of entrance fees and dues		$17,185.00
Sales of publications		2,182.25
Banquet, annual meeting, 1918		3,745.00
Banquet, annual meeting, 1919, reservations		1,191.00
Exchange on remittances of members		4.30
Miscellaneous receipts		30.00
Interest received:—		
On bank balances	$69.80	
On investments—other than endowment fund	538.95	
Endowment fund—municipal bonds	490.00	
Endowment fund—Liberty Bonds	343.31	
		1,442.06
Members' contribution to endowment fund		2,550.00
Total receipts		$28,329.61
Balance—Cash in bank and on hand, November 1, 1918		4,953.47
		$33,283.08

STATEMENT.

DISBURSEMENTS.

From November 1, 1918, to October 31, 1919, inclusive.

Publications:—		
Preliminary papers, 1918	$4,063.68	
Volume 26, 1918, proceedings	7,858.55	
1919 Year Book	719.78	
Miscellaneous printing	194.10	
Storage of volumes	232.62	
Forwarding volumes and papers	107.12	
		$13,175.85
Expenses of 26th annual meeting, 1918		1,474.86
Expenses of 26th annual banquet, 1918		3,809.85
Expenses of Philadelphia meeting, 1918, Entertainment Committee		2,208.50
Rent of Society rooms		294.00
Salaries of officers and clerk		2,131.62
Office expenses, postage, stationery, etc.		1,410.85
Audit of books and accounts year ending October 31, 1918		285.00
Exchange on remittances of members		20.82
Trustees' commissions—Franklin Trust Company		25.73
Investments in Liberty and Victory Bonds		1,960.00
Accrued interest on above bonds		18.87
Refund of H. L. Aldrich prize		100.00
Endowment fund investment in Liberty Bonds, $2,700, par value		2,547.22
Total disbursements		$29,463.17
Balance, October 31, 1919:—		
On deposit with Franklin Trust Company	$3,406.37	
Checks undeposited	40.00	
Cash on hand	27.45	
Total general surplus cash	$3,473.82	
Endowment fund cash on deposit with Franklin Trust Company, Trustee	346.09	
		3,819.91
		$33,283.08

DEATHS (11).

Members (7).

Albert Allen.
Edward P. Bates.
Marshall T. Davidson.
William D. Dickey.
Richard L. Newman.
Rudolph Veith.
John M. Williamson.

Associates (3).

Clement A. Griscom, Jr.
William H. Pleasants.
Richard C. Veit.

Junior (1).

Parr Hooper.

RESIGNATIONS (24).

Members (13).

Ludwig Anderssen.
Cornelius A. Binks.
C. D. Bray.
Arthur B. Cassidy.
John M. Cherry.
William Cowie.
James A. Davies.
Edwin O. Fitch.
Ralph T. Hanson.
David I. Irish.
James McFarlane.
Carlo Pfister.
St. Clair T. Thomas.

Associates (11).

Harry S. Bradley.
Harry S. Demarest.
Edward P. Field.
Elihu B. Frost.
Edward R. Grabow.
Theodore E. Hammond.
Frederick Holbrook.
Frank B. Jones.
Louis Kempff.
Howard L. Shaw.
Horace S. Wilkinson.

Statement of Resources and Liabilities as at October 31, 1919.

RESOURCES.

Cash in bank and on hand		$3,473.82
Membership dues and entrance fees (after deducting uncollectible accounts), per Schedule "1"		2,905.00
Due from dealers and members for publications		755.50
Publications on hand:—		
Volumes Nos. 1 to 18, inclusive, at 50 per cent of cost; Volumes Nos. 19 to 26, inclusive, at cost		7,304.44
Office furniture and equipment, at 30 per cent of cost		885.00
Investments:—		
$4,000 Baltimore & Ohio R. R. Conv. 4½% bonds, at cost	$3,925.00	
$3,000 Central Pacific Ry. 1st Ref. 4% bonds, at cost	2,651.25	
$2,000 Morris & Essex R. R. 1st Ref. 3½% bonds, at cost	1,732.50	
$5,000 U. S. Liberty and Victory Bonds, at par	5,000.00	
		13,308.75

Endowment Fund:—		
Investments at par:		
New York City 4½% bonds, due May 1, 1957	$7,000.00	
Brooklyn Public Market 3½% bonds	5,000.00	
U. S. Liberty Bonds	10,050.00	
	$22,050.00	
Cash on deposit with trustee	346.09	
		22,396.09
Total resources		$51,028.60

LIABILITIES.

Annual banquet, 1919, reservations		$1,191.00
Entrance fees and dues paid in advance		190.00
Total liabilities		$1,381.00
Society's net worth at October 31, 1919:—		
Endowment fund, per Schedule "2"	$33,161.09	
Net worth other than endowment fund, per Schedule "3"	16,486.51	
		49,647.60
		$51,028.60

SCHEDULE "1."

Membership Dues and Fees Delinquent as at October 31, 1919.

Particulars	*Members*	*Associates*	*Juniors*	*Total*
Dues for year 1915	$10.00	......	......	$10.00
1916	50.00	$20.00	......	70.00
1917	160.00	45.00	$10.00	215.00
1918	310.00	100.00	10.00	420.00
1919	1,180.00	550.00	65.00	1,795.00
Entrance fees	200.00	180.00	15.00	395.00
	$1,910.00	$895.00	$100.00	$2,905.00

The amount due as above, $2,905.00, is due from Active Members only, all known deaths, resignations and uncollectible accounts having been written off, as per Schedule "3."

SCHEDULE "2."

Endowment Fund as at October 31, 1919.

Particulars		*Amount*
Balance, November 1, 1918		$17,980.00
Contributions by members, in:—		
Liberty Bonds at par	$7,350.00	
Cash	2,550.00	
		9,900.00

Purchases of Liberty Bonds at a Discount:			
Particulars			*Amount*
Purchased at par value		$2,700.00	
Cost of $2,700 bonds		2,547.22	
			$152.78
Interest earned on:—			
Liberty Bonds contributed and purchased		$343.31	
$12,000 Municipal bonds		490.00	
			833.31
Membership fees collected:—			
Entrance fees of new members		$4,095.00	
Life membership paid in cash	$160.00		
Transfer from dues paid in advance as November 1, 1918, to apply to life membership	40.00		
		200.00	
			4,295.00
Total, October 31, 1919			$33,161.09

SCHEDULE "3."

Net Worth of Society other than Endowment Fund as at October 31, 1919.

Balance, November 1, 1918			$17,187.38
Additions:—			
Dues for year 1919		$13,735.00	
Dues of members reinstated		110.00	
		$13,845.00	
Less dues written off:—			
Deaths and resignations	$135.00		
Non-payment	280.00		
Suspended	50.00		
Failure to qualify	10.00		
		475.00	
Net dues receivable for year			13,370.00
Membership entrance fees for year		$4,850.00	
Less: Failure to qualify	$10.00		
Collected fees transferred to endowment fund, per Schedule "2"	4,295.00		
		4,305.00	
Fees not collected during the year			545.00

Sales of publications		$2,372.86
Surplus on annual banquet		395.15
Surplus on Philadelphia meeting, Entertainment Committee		191.50
Miscellaneous receipts		35.00
Interest earned:—		
On investments (other than endowment fund)	$520.08	
On bank balance	69.80	
		589.88
Liberty bonds purchased in 1918 at a discount		50.00
Total		$34,736.77
Deductions:—		
Cost of publications distributed to members and sold to dealers	$12,276.04	
Expenses of 26th annual meeting	1,474.86	
Rent of Society rooms	294.00	
Salaries of officers and clerk	2,131.62	
Office expenses, postage, stationery, etc.	1,410.85	
Audit of accounts for year ended October 31, 1918	285.00	
Exchange on remittances of members	16.52	
Transportation of volumes	93.62	
Storage of volumes	232.62	
Trustees' commissions	25.73	
Uncollectible accounts for publications—written off	9.40	
		18,250.26
Balance, October 31, 1919		$16,486.51

The financial statements and books of account have been audited by Mills & Company, Public Accountants, 42 Broadway, New York, N. Y.

Respectfully submitted,

DANIEL H. COX,

Secretary-Treasurer.

SECRETARY-TREASURER COX:—Mr. President and gentlemen, the report, as usual, has been printed and distributed. I must ask your indulgence for certain typographical errors and errors that will be corrected in the final report. The printers, unfortunately, are following the prevailing custom of making difficulties. I will touch on some of the points contained in the report briefly.

It is naturally very pleasant to note that the membership at this time is 1,386, as against a membership at the corresponding time last year of 960. We made a very active campaign two years ago for new members and it was quite successful. During the past year we have not had quite as good an opportunity to induce new members to come in, because the year before there were so many new people interested in shipbuilding, and shipyards generally, that it was a very opportune time to get new members for our Society. We have, however, at this time a large number of applicants for membership in the Society which have been passed on by the Council and will be presented in due time. But the opinion of the Council is that the membership at large should take during the coming year a very direct interest in recruiting from proper material new members for the Society. The larger the membership of the Society is, the more papers of value we will be able to procure and the greater the value of the Society in general will be, so it is hoped that all members will from time to time try to interest eligible men to take an interest in and become members of the Society. In that connection I will say that it is not too late to have applications for membership submitted at this meeting and acted upon, and there is a supply of the blank forms in the rear of the hall, so if you think of any possible members that you have not already talked to, you might approach them, and, if possible, secure their membership during the next two days.

The financial statement is in quite complete detail. You will be interested to see that the Endowment Fund has now reached a very sizable proportion. Its present value is in excess of $33,000. That at the time of coming to the meeting was the book value and is partly represented by investments already placed in the Endowment Fund and partly represented by other investments and cash in hand. At the meeting of the Council it was arranged to transfer to the Endowment Fund securities to the total amount of $33,161.09, so that we will have securities on hand to that amount in the Endowment Fund.

Receipts and disbursements during the year upon a final analysis, after making the additions to the Endowment Fund, which were provided for by the action taken some time ago, show a deficit in operation for the year of approximately $700. That is accounted for directly by two matters which are easily explained.

In the first place, the meeting last year was held in Philadelphia, and notwithstanding the very generous contributions of the Philadelphia membership and others towards the expenses of the meeting, there were certain additional expenses as a result of holding the meeting in Philadelphia, and not in New York; and, secondly, the volume of Proceedings for the last year was a very large one—a very interesting one, but extremely large—and it had a great number of very expensive illustrations which made the cost of publication very high.

While, therefore, we have theoretically run behind to a slight extent, this is accounted for directly by the unusual expenditures referred to which it is hoped we will not have to meet this year, as there is every indication that the Society is on a strong and satisfactory basis. I think the report, as presented, with these explanations is quite intelligible to the membership.

THE PRESIDENT:—Will you please rise, gentlemen, during the reading of the list of those who have died during the year?

The Secretary then read the list of the deceased Members and Associates.

The President:—Among those who have died during the past year appear the names of three who have been members of the Society from its organization—Messrs. Davidson, Dickey and Veit. They gave to the Society the benefit of their rich experience, and we shall miss their presence very much.

In the list of Associates appears the name of Clement A. Griscom, Jr., son of our lamented first president, and himself one of our earliest and most valuable members.

Another on the list, although only a Junior Member, has shed a special luster upon the Society, for Lieut. Parr Hooper answered his country's call and made the supreme sacrifice on the fields of France.

Others of those whose names have just been read by the secretary have entered fully into the activities of this Society, among them Mr. Bates, who was one of our oldest members, Messrs. Pleasants, Allen, Newman, and Williamson.

The passing of all of these members we note with the deepest regret.

The secretary will now read the list of applicants who have been recommended by the Council for election to membership in this Society.

The Secretary:—The total number of applicants who have applied for the grade of full Member and have been recommended favorably by the Council is 96, the names being as follows:

Members (96).

Sir Westcott S. Abell Chief Hull Surveyor, Lloyd's Registry of Shipping, 11 Fenchurch Street, London, England.

James Scott Abrams, Cox and Stevens. P. O. address: 15 William Street, New York, N. Y.

Maxwell Alpern, Vice-President and General Manager, American Engineering Co., Philadelphia, Pa.

Wm. P. Arrott, Surveyor, American Bureau of Shipping, 66 Beaver Street, New York, N. Y.

William T. Bonner, Engineer Welding Research Department, New York Shipbuilding Corporation. P. O. address: c/o New York Shipbuilding Corporation, Camden, N. J.

Wm. L. Bunker, Chief Inspector, Construction and Repair Department, New York District, U. S. S. B., Division of Operations, 45 Broadway, New York, N. Y. P. O. address: 129 Columbia Heights, Brooklyn, N. Y.

Richard L. Burke, Chief Draughtsman, Sun Shipbuilding Co., Chester, Pa. P. O. address: 1417 Melrose Avenue, Chester, Pa.

Clarence Leslie Burns, Chief Estimator, Bethlehem Shipbuilding Corporation, Ltd., Bethlehem, Pa. P. O. address: 622 West Market Street, Bethlehem, Pa.

Thomas Cameron, American Bureau of Shipping, 66 Beaver Street, New York, N. Y. P. O. address: 589 Ridgewood Road, Maplewood, N. J.

James A. Capstaff, Chief Engineer and Plant Engineer, W. T. A. Fletcher Company, Hoboken, N. J. P. O. address: S. E. corner Palisade Street and Lafayette Avenue, Grantwood, N. J.

Horace N. Chippendale, Resident Inspector, Emergency Fleet Corporation. P. O. address: 702 S. Webster Avenue, Green Bay, Wis.

Arthur J. Coughtry, Estimator, Bethlehem Shipbuilding Corporation, Bethlehem, Pa.

Percival R. Court, Engineer Surveyor, British Corporation for the Survey and Registry of Shipping. P. O. address: In care of British Corporation, Board of Trade Building, Toronto, Ontario, Canada.

David Currier, Jr., Engineer and Purchasing Agent, Norway Pacific Construction & Dry Dock Company, Everett, Wash. P. O. address: 1627 Lamont Street, Washington, D. C.

John F. Dick, Naval Architect, Columbia River Shipbuilding Corporation, Portland, Ore.

Wayne T. Dimm, Assistant Chief Engineer, Newport News Shipbuilding and Dry Dock Co., Newport News, Va.

Wm. S. Doran, President, Alberger Pump and Condenser Company, 140 Cedar Street, New York, N. Y.

Alfred Dunn, Superintendent Engineer, Isthmian S.S. Lines, 11 Broadway, New York, N. Y.

Chas. W. Dyson, Rear Admiral, U. S. N , Head of Design Division, Bureau of Steam Engineering, Navy Department, Washington, D. C. P. O. address: 1840 Lamont Street, Washington, D. C.

Robt. Wm. Erickson, Chief Electrical Draughtsman, Federal Shipbuilding Company, Kearney, N. J. P. O. address: 114 Bartholdi Avenue, Jersey City, N. J.

Frank E. Ferris, Managing Agent, U. S. Shipping Board. In charge of Construction and Repair Department of the New York District. P. O. address: 45 Broadway, New York, N. Y.

Paul Foley, Captain, U. S. N., Chartering and Tank Steamer Executive, U. S. Shipping Board, New York, N. Y. P. O. address: No. 2 Gramercy Park, New York, N. Y.

John Forsyth, Marine Superintendent, New England Fuel & Transportation Co., 111 Devonshire Street, Boston. P. O. address: 22 Asticon Road, Forest Hills, Boston.

Albert Frankish, Assistant to Machinery Engineer, Hog Island, Pa. P. O. address: Box 96, Post Office, Gloucester, N. J.

Truman A. Gamon, Plant Manager, Welin Plant, American Balsa Co., Long Island City, N. Y. P. O. address: 139 25th Street, Elmhurst, N. Y.

Harry E. D. Gould, General Superintendent, Fore River Works, Bethlehem Shipbuilding Corporation. P. O. address: 58 Greenleaf Street, Quincy, Mass.

Wm. I. Hay, Surveyor to the British Corporation for the Survey and Registry of Shipping, 14 Blythwood Square, Glasgow, Scotland. P. O. address: British Corporation for Survey and Registry of Shipping, Board of Trade Bldg., Toronto, Canada.

Leonard Bushe Harris, Consulting Engineer and Naval Architect. P. O. address: Woolworth Bldg., New York, N. Y.

Jos. Hecking, Engineer Surveyor, American Bureau of Shipping, 66 Beaver Street, New York, N. Y. P. O. address: 94 Wheaton Place, Rutherford, N. J.

Geo. Heatherton, General Manager, Sparrow's Point Plant of Bethlehem Shipbuilding Corporation, Ltd., Sparrow's Point, Md.

Clyde Howard, General Manager and Treasurer, Howard Ship Yard and Dock Co., Jeffersonville, Ind.

Cyril P. Hubert, Chief Estimator, Los Angeles Shipbuilding and Dry Dock Company. P. O. address: 1263 West 38th Street, Los Angeles, Cal.

J. Wm. Hudson, Chief Draughtsman, Sun Shipbuilding Company. P. O. address: 518 E. 21st Street, Chester, Pa.

Robt. Jacob, Jr., In charge, Passenger Vessel Section, Construction and Repair Department, U. S. Shipping Board, 45 Broadway, New York, N. Y. P. O. address: City Island, N. Y.

Geo. Jenkins, Assistant to Engineer (Engineering Division), A. I. S. C., Hog Island, Pa. P. O. address: 225 S. 40th Street, Philadelphia, Pa.

Sidney Grant Jenks, General Superintendent, South Yard, New York Shipbuilding Corporation, Camden, N. J. P. O. address: Haddonfield, N. J.

Willard Francis Jones, Assistant Marine Superintendent. P. O. address: 24 State Street, New York, N. Y.

P. W. Keltie, Assistant to Mr. Geo. B. Ward, Chief Inspector, E. F. C. Office, Cleveland, Ohio. P. O. address: 206 The Fortieth, 2029 E. 40th Street, Cleveland, Ohio.

Geo. E. Lawrence, Sales Manager, Redington Standard Fittings Company, Bethlehem, Pa.

Francis W. Leahy, Senior Performance Engineer, U. S. Shipping Board, Emergency Fleet Corporation. P. O. address: Apt. 5, Florence Court, 47 Pierrepont Street, Brooklyn, N. Y.

John Lockhart, General Manager, J. Coughlan & Sons, Shipbuilders and Engineers, Vancouver, B. C.

John Lyell, Resident Inspector, Emergency Fleet Corporation. P. O. address: 2160 East 40th Street, Cleveland, Ohio.

Eben R. Macmillan, Surveyor to the British Corporation for the Survey and Registry of Shipping, 14 Blytheswood Square, Glasgow, Scotland. P. O. address: British Corporation for Survey and Registry of Shipping, Board of Trade Building, Toronto, Canada.

Louis P. Maier, Engineering Superintendent, Kensington Shipyard Department of the Wm. Cramp & Sons Ship and Engine Building Company, Beach and Palmer Streets, Philadelphia. P. O. address: 2511 N. 30th Street, Philadelphia, Pa.

Matthew M. Marshall, Assistant Chief Inspector, U. S. Shipping Board, 45 Broadway, New York, N. Y.

Geo. R. Martin, Senior Performance Engineer, U. S. Shipping Board, Custom House, Boston, Mass. P. O. address: 13 Liberty Street, Peabody, Mass.

Harry S. Marvel, Tank Shipbuilding Company, Newburgh, N. Y.

Earl Potter Mason, Commander, U. S. N. (Reserve Force). P. O. address: 2 and 4 Stone Street, New York, N. Y.

Harold R. McClelland, Member of the firm of N. E. McClelland & Company. P. O. address: Imperial Bank Chambers, 286 St. James Street, Montreal, Canada.

Eugene F. McDermott, Jr., Technical Assistant, District Manager's Office, Delaware River District, Emergency Fleet Corporation, 140 N. Broad Street, Philadelphia, Pa. P. O. address: 215 S. 50th Street, Philadelphia, Pa.

Joseph M. McDermott, Assistant Chief Draughtsman, A. I. S. C., Hog Island, Pa. P. O. address: 19 New Street, Upper Darby, Delaware Co., Pa.

Irving Metten, Assistant Material Officer, N. O. T. S., Third N. D. P. O. address: 215 West 88th Street, New York, N. Y.

David S. Miller, Marine Superintendent, Cunard Steamship Company, New York, N. Y. Cunard Pier 54, N. R.

Wm. M. Mills, General Manager, Kingston Shipbuilding Corporation, Kingston, N. Y. P. O. address: 260 Clinton Avenue, Kingston, N. Y.

Peter Mitchell, Naval Architect and Chief Draughtsman, Standard Shipbuilding Corporation, Shooters Island, N. Y. P. O. address: 49 Fairview Avenue, West New Brighton, Staten Island, N. Y.

A. B. Morrissey, District Representative on Troop Ships, Delaware River District, U. S. S. B., E. F. C., 140 N. Broad Street, Philadelphia, Pa. P. O. address: 1241 W. Allegheny Avenue, Philadelphia, Pa.

Walter Muller, Chief Draughtsman, LeParmentier Shipbuilding Company, 43d Floor, Woolworth Bldg., New York, N. Y. P. O. address: 43d Floor, Woolworth Bldg., New York, N. Y.

Harry G. Munro, Assistant Hull Fittings Engineer, A. I. S. C., Hog Island, Pa. P. O. address: 5627 Washington Avenue, Philadelphia, Pa.

Richard Mussen, Engineer Surveyor to the American Bureau of Shipping, Portland, Ore. P. O. address: 837 Northwestern Bank Bldg., Portland, Ore.

Octavius Narbeth, Principal Surveyor, Lloyd's Register of Shipping. P. O. address: 449-50-51 Bourse Bldg., Philadelphia, Pa.

Edw. A. Oppelt, Resident Inspector, Emergency Fleet Corporation, New York Shipbuilding Corporation, Camden, N. J. P. O. address: N. Y. Shipbuilding Corporation, Camden, N. J.

John I. Palmer, Chief Hull Fittings Draughtsman, Hog Island, Pa. P. O. address: 4209 Baltimore Avenue, Philadelphia, Pa.

Roy F. Parris, Chief Inspector, Delaware River District, E. F. C., U. S. S. B., 140 N. Broad Street, Philadelphia, Pa.

Harry Garfield Peake, Vice-President and General Manager, Union Construction Company. P. O. address: 244 16th Avenue, San Francisco, Cal.

Chas. B. Phillips, Superintendent, Atlantic Basin Iron Works, 168 Van Brunt Street, Brooklyn, N. Y.

William Harvey Phillips, Superintending Engineer, U. S. Transport Service, Room No. 425, Pier No. 3, Hoboken, N. J. P. O. address: Masonic Cub, New York, N. Y.

A. F. Pillsbury, District Manager, E. F. C., Southern Pacific District. P. O. address: 1547 Spruce Street, Berkeley, Cal.

Chas. L. Putzel, Chief Hull Draughtsman for Ames Shipbuilding and Dry Dock Company, Seattle, Wash.

Henry E. Quenstedt, Marine Superintendent, Carolina Company, Charleston, S. C.

John A. Raidabaugh, Assistant Engineer, A. I. S. C., Hog Island, Pa. P. O. address: Powelton Apartment, 35th Street and Powelton Avenue, Philadelphia, Pa.

Lincoln DeBreuner Randall, Assistant Chief Hull Draughtsman, Federal Shipbuilding Company, Kearney, N. J. P. O. address: 9 Watson Avenue, Newark, N. J.

Fred J. Ruch, Production Engineer, Newburgh Shipyards, Inc., Newburgh, N. Y. P. O. address: 158 Dubois Street, Newburgh, N. Y.

Wm. E. Russell, Gear and Turbine Expert, U. S. S. B., 45 Broadway, New York, N. Y. P. O. address: 477 St. Johns Place, Brooklyn, N. Y.

Russell E. Stabler, Chief Machinery Draughtsman, A. I. S. C., Hog Island, Pa. P. O address: 703 E. E. Street, Sparrows Point, Md.

H. E. Saunders, Lieutenant Commander (C. C.), U. S. N., Office and Planning Superintendent, Hull Division, Navy Yard, Mare Island, Cal.

Wm. V. Sauter, Mechanical Engineer, American Engineering Company, Aramingo Avenue and Cumberland Street, Philadelphia, Pa. P. O. address: 2305 N. Broad Street, Philadelphia, Pa.

Walter Scott, Supervising Superintendent, Engineer Division of Operations, U. S. S. B., Custom House, N. Y. P. O. address: 213 79th Street, Brooklyn, N. Y.

Alexander Schein, Engineer in Charge of Ship Stabilizing Department, Sperry Gyroscope Company, Brooklyn, N. Y. P. O. address: 476 Clinton Avenue, Brooklyn, N. Y.

Wm. C. Shaw, Chief Engineer, Columbia River Shipbuilding Corporation, Portland, Ore. P. O. address: 403 East 40th Street, North, Portland, Ore.

Jos. E. Sheedy, Assistant to the President, Downey Shipbuilding Corporation, Arlington, Staten Island, N. Y. P. O. address: 218 Kissel Avenue, New Brighton, Staten Island, N. Y.

William E. Sizemore, Surveyor, American Bureau of Shipping, Securities Bldg., Seattle, Wash.

Francis H. Short, Superintendent of Hull Construction, Nashville Bridge Company, Nashville, Tenn. P. O. address: 207 Gallatin Road, Nashville, Tenn.

Henry Orlando Slayton, Surveyor, American Bureau of Shipping, 66 Beaver Street, New York, N. Y.

Frederic E. Smith, Mechanical Designer, Pensacola Shipbuilding Company, Pensacola, Fla.

Wm. G. Smith, General Manager, Anchor Shipbuilding Corporation, Washburn, Wis.

John Strachan, General Superintendent, Machinery Installation, A. I. S. C., Hog Island, Pa. P. O. address: 48 Glendale Road, Upper Darby, Pa.

Christopher Story, Jr., Manager, Marine Department, Henry L. Doherty & Company, 60 Wall Street, New York, N. Y. P. O. address: Room 1201, 60 Wall Street, New York, N. Y.

Paul A. Talbot, Surveyor, American Bureau of Shipping, 66 Beaver Street, New York, N. Y.

Georgory Isaac Touriansky, Assistant Chief Draughtsman, Electrical Section, A. I. S. C., Hog Island, Pa. P. O. address: 5014 Wayne Avenue, Germantown, Philadelphia, Pa.

Henry R. Trotter, Chief Engineer, Ball Bearing Company, Hartford, Conn. P. O. address: 187 Park Road, Hartford, Conn.

Lawrence A. Walker, Surveyor, American Bureau of Shipping, 66 Beaver Street, New York, N. Y. P. O. address: 638 52nd Street, Brooklyn, N. Y.

William McGehee Wallace, Civil Engineer, Rensselaer Polytechnic Institute. P. O. address: Bureau Construction and Repair, Navy Department, Washington, D. C.

Albert Edward Wilson, Inspector, Gulf District for Bureau Veritas, Consulting Engineer, Cuyamel Fruit Company, New Orleans, La. P. O. address: 301 Title Guarantee Bldg., New Orleans, La.

Julius Adolph Woidill, Jr., Electrical Chief Draughtsman, A. I. S. C., Hog Island, Pa. P. O. address: 513 Atlantic Avenue, Egg Harbor City, N. J.

Henry C. Wright, Assistant Chief Hull Draughtsman, Bath Iron Works, Ltd., Bath, Me. P. O. address: Elmhurst, Bath, Me.

Jos. T. Yates, Superintendent of Lighthouses, Lighthouse Department, Tompkinsville, Staten Island, N. Y.

THE PRESIDENT:—Gentlemen, you have heard the list of those recommended by the Council for membership in The Society of Naval Architects and Marine Engineers. What is your pleasure with respect thereto?

MR. STEVENSON TAYLOR, *Past President:*—I move that the recommendation of the Council be approved, and the applicants whose names have been read by the secretary be elected as Members of the Society.

The motion was duly seconded, put to vote and carried.

THE SECRETARY:—The list of applicants for the grade of Associate Member, which has been favorably recommended by the Council, comprises forty-five names and is as follows:—

Associates (45).

Arthur Frederick Aldridge, Editor, *The Rudder,* President, the Rudder Publishing Company. P. O. address: 9 Murray Street, New York, N. Y.

Geo. W. Betts, Member of Hunt, Hill & Betts, Proctors in Admiralty, 120 Broadway, New York, N. Y.

Paul H. Blair, Boat Foreman, American Shipbuilding Company, Lorain, Ohio. P. O. address: 1121 Sixth Street, Lorain, Ohio.

Leo S. Blodgett, Assistant Editor, *Marine Engineering,* 6 E. 39th Street, New York, N. Y. P. O. address: 33 Park Street, Montclair, N. J.

Joseph Edward Buckley, Chargeman, Machinery Draughting Division, A. I. S. C., Hog Island, Pa. P. O. address: 2610 S. 70th Street, West Philadelphia, Pa.

Roland P. Carr, Lieutenant J. G. (C. C.), U. S. Navy Yard, N. Y.

Piero Civalleri, Captain, Royal Italian Navy, Naval Attaché to the Royal Italian Embassy. P. O. address: 1752 N Street, N. W., Washington, D. C.

John W. Crandall, Lawyer, Member of firm of Hunt, Hill & Betts, 120 Broadway, New York, N. Y.

Jerome B. F. Crowley, Statistician, American Bureau of Shipping. P. O. address: 628 W. 158th Street, New York, N. Y.

John Henry Dialogue, Jr., Marine Engineer and Piping Draughtsman, 1st Class, The Atlantic Corporation, Portsmouth, N. H. P. O. address: 40 Saratoga Way, Portsmouth, N. H.

John A. Donald, Commissioner, U. S. S. B., Vice-President, Emergency Fleet Corporation. P. O. address: 1319 F Street, Washington, D. C.

Chas. E. Fraser, President, Fraser Brau & Clarke Dry Dock Corporation, N. Y.; President Fraser Brau Shipyards, Ltd., Montreal. P. O. address: 1328 Broadway, New York, N. Y.

Arnljot Frederick Christian Grontoft, District Statistician, Delaware River District, U. S. S. B., E. F. Corporation, Philadelphia, Pa. P. O. address: 1402 Market Street, Wilmington, Del.

Adolph A. Gathemann, Lieutenant, U. S. N. P. O. address: 12 Warwick Road, Brookline, Mass.

R. J. Hall, Assistant to Chief Draughtsman, Sun Shipbuilding Company, Chester, Pa. P. O. address: 5919 Irving Street, Philadelphia, Pa.

Shuzo Inaba, Hull Draughtsman, Newburgh Shipyard. P. O. address: 181 Prospect Street, Newburgh, N. Y.

Harry Leon Katz, Draughtsman, Hull Division, Scientific Department, Union Shipbuilding Company, Baltimore, Md. P. O. address: 925 Harlem Avenue, Baltimore, Md.

Teiji Kawamura, Representative in New York of the Osaka Iron Works, Ltd. P. O. address: Osaka Iron Works, Ltd., 26 Cortlandt Street, New York, N. Y.

Wm. Edward Kennedy, Manager of the Shipbuilding Cyclopedia, Woolworth Bldg., New York, N. Y.

Clinton B. Kolyer, Draughtsman, Los Angeles Shipbuilding and Dry Dock Company, San Pedro, Cal. P. O. address: 3015 Theresa Street, Long Beach, Cal.

Wm. A. Lake, Pantasote Co., 11 Broadway, New York, N. Y.

Thomas Charles Landi, member of firm of Cox and Stevens, 15 William Street, New York, N. Y. P. O. address: 15 William Street, New York, N. Y.

Lester Hayes Laraway, Hull Draughtsman, Los Angeles Shipbuilding and Dry Dock Company, San Pedro, Cal. P. O. address: 746 E. Tenth Street, Long Beach, Cal.

John Harrar Levy, Assistant Chief Draughtsman (Machinery), A. I. S. C., Hog Island, Pa. P. O. address: 5128 N. 15th Street, Philadelphia, Pa.

Clifford Day Mallory, President, C. D. Mallory & Company, Inc., and Baltimore Oceanic Steamship Company. P. O. address: 10 Bridge Street, New York, N. Y.

John W. Mason, President, Western Pipe and Steel Company of California. P. O. address: 444 Market Street, San Francisco, Cal.

Armand Mayville, Lieutenant (J. G.), C. C., U. S. N., Naval Station, New Orleans, La. P. O. address: Warrington, Fla.

Wm. H. Milne, Apprenticed as Ship Draughtsman with the Fairfield Shipbuilding Company, Glasgow, Scotland, August, 1912, to January, 1913. P. O. address: c/o M. E. McClelland & Co., Ltd., Imperial Bank Chambers, 286 St. James Street, Montreal, Canada.

Joseph A. Moore, Vice-President and General Superintendent, Moore Shipbuilding Company. P. O. address: Moore Shipbuilding Company, Oakland, Cal.

Rupert MacConnell Much, Sr., Assistant Plant Manager, Virginia Shipbuilding Corporation, Alexandria, Va. P. O. address: 124 Walnut Street, Alexandria, Va.

Lynn W. Nones, Manager, Marine Department, Griscom-Russell Co., 90 West Street, New York, N. Y.

Otho M. Otte, General Manager and Engineer, Interior Metal Manufacturing Company. P. O. address: Brackenridge, Pa.

Fernand-Roger de Perrot, Leadingman Shipfitter, Bethlehem Shipbuilding Corporation, Fore River Plant. P. O. address: 63 Adams Street, Quincy, Mass.

W. S. Rugg, Manager, Marine Department, Westinghouse Electric and Manufacturing Company, East Pittsburgh, Pa. P. O. address: University Club, Pittsburgh, Pa.

John Sampson, Vice-President, Oriental Navigation Company. P. O. address: 17 Battery Place, New York, N. Y.

Adolph Carl Alexander Sandner, Hull Inspector, U. S. Shipping Board, E. F. C., c/o Los Angeles Shipbuilding and Dry Dock Company, San Pedro, Cal. P. O. address: 1334 Bonita Avenue, Berkeley, Cal.

Jos. Louis Silverman, Hull Inspector, E. F. C., Standard Shipbuilding Corporation. P. O. address: 35 Buffalo Avenue, Brooklyn, N. Y.

Ernst Smith, Manager, Steamship Department, Vacuum Oil Company, 61 Broadway, New York, N. Y. P. O. address: 642 E. 15th Street, Brooklyn, N. Y.

Geo. C. Sprague, Member of firm of Hunt, Hill & Betts, 120 Broadway, New York, N. Y. P. O. address: Englewood, N. J.

Arthur Wendel Stout, Assistant to General Superintendent, Alabama Dry Dock and Shipbuilding Company. P. O. address: 927 Dauphin Street, Mobile, Ala.

David Todd Warden, Manager, Foreign Shipping Department, Standard Oil Co. (N. J.). P. O. address: 26 Broadway, New York, N. Y.

E. L. Warner, Manager, Marine Department, Detroit Graphite Company, Eastern Division, Equitable Bldg., New York, N. Y. P. O. address: 320 Seneca Avenue, Mt. Vernon, N. Y.

The Secretary:—The following applicants for the grade of Junior Member have been received and recommended for favorable action, nine in all.

Juniors (9).

Ray Eber Brown, Lieutenant (J. G.), C. C., U. S. Navy, Inspection Duty, New York Shipbuilding Corporation, Camden, N. J. P. O. address: 5442 Sansom Street, Philadelphia, Pa.

Lamar P. Harrison, Estimator, U. S. S. B., E. F. C. P. O. address: 809 Hibernia Bank Bldg., New Orleans, La.

K. N. Luks, Technical Staff, American Bureau of Shipping, New York, N. Y. P. O. address: Towaco, N. J.

Mason S. Noyes, Draughtsman, Union Shipbuilding Company, Baltimore, Md. P. O. address: 11 West Franklin Street, Baltimore, Md.

Geo. A. Smith, Technical Staff, American Bureau of Shipping, 66 Beaver Street, New York, N. Y. P. O. address: 14 W. 60th Street, New York, N. Y.

C. H. Stevens, Draughtsman, Standard Oil Co. of New York, 26 Broadway, New York, N. Y. P. O. address: 172 W. 105th Street, New York, N. Y.

C. H. Young, Surveyor with American Bureau of Shipping, 66 Beaver Street, New York, N. Y. P. O. address: 1 West 127th Street, New York, N. Y.

Kenneth Warren Heinrich, Superintendent, Cancellation Section, Estimating Division, Bethlehem Shipbuilding Corporation, Ltd. P.O. address: Bethlehem Club, Bethlehem, Pa.

Edward Maurice Kent, Estimator and Calculator in Scientific Section of the Lake Torpedo Boat Company, Bridgeport, Conn. P. O. address: 1034 Howard Avenue, Bridgeport, Conn.

The Secretary:—There are in addition a certain number of transfers of members from one grade to another which will be announced later in the morning.

The President:—Gentlemen, you have heard the list of applicants for Associate Membership and Junior Membership, as approved by the Council. What is your pleasure?

Captain J. H. Linnard, C. C., U. S. N., *Member of Council:*—I move that the action of the Council with respect to the election of Associate Members and Junior Members be approved.

The motion was duly seconded, put to vote and carried.

The President:—Gentlemen, in accordance with the provisions of the Constitution and By-laws the following officers were elected at a meeting of the Council held yesterday afternoon:

For Vice-Presidents, term expiring December 31, 1922:—A. P. Niblack, R. M. Watt, H. D. Goulder, C. P. Wetherbee.

For Members of Council, term expiring December 31, 1922:—J. H. Linnard, C. A. McAllister, W. L. R. Emmet, J. H. Gardner, W. J. Davidson, H. P. Frear.

For Associate Members of Council, term expiring December 31, 1922:—R. A. C. Smith, A. W. Goodrich.

For Executive Committee:—Stevenson Taylor, Andrew Fletcher, W. M. McFarland, F. L. DuBosque, J. W. Powell.

For Secretary-Treasurer:—D. H. Cox.

For Committee on Papers:—W. M. McFarland, F. L. DuBosque, H. L. Aldrich.

At this stage of the proceedings, it has been customary for the president to make a few remarks.

President Capps then read the following address:—

PRESIDENT'S ADDRESS

It has been customary in the past for the president of this Society to make some introductory remarks before proceeding to the reading of technical papers presented for your consideration. These remarks have most frequently taken the form of a brief review of progress in shipbuilding and marine engineering during the preceding year.

These addresses have been in great part rich in statistical data and general information concerning the progress of the art whose development is the primary reason for our existence as a Society.

In preparation for this part of his duties, your president has collected from time to time some interesting professional data. Much of it, however, was obtained in a form which up to the present time has not permitted its release for publication. Under the circumstances, therefore, there will be no attempt to present to you at this time any comprehensive review of conditions in the shipbuilding world.

Although it is not my purpose to do more than merely touch upon our shipbuilding development, it may be of interest to compare briefly conditions of today with those noted in three previous presidential addresses. In November, 1900, President Griscom made the following statement:—

"The most prosperous year shipbuilding in the United States has known since the outbreak of the Civil War nears its end.

"The new year and the century about to begin bid fair to witness a development of the industry responsive to our high hopes at the time the Society of Naval Architects and Marine Engineers was founded.

"During the fiscal year of the Government, which ended in June, eighty steel steam vessels of 167,948 gross tons were built in the United States. These figures are modest compared with Great Britain's output of 567 steel steam vessels of 1,341,425 gross tons during the year; but they are full of encouragement when put beside the fact that during the previous nine years the United States built only 574,802 gross tons of these types."

Again, in 1908, Admiral Bowles, as President of the Society, commented upon the adverse effect upon shipbuilding of the "general depression that has weighed upon all other business interests," and then went on to say:—

"Because of orders placed before this depression began, the shipbuilding output of the fiscal year ending June 30, 1908, was the greatest in our history, attaining a total of 1,457 vessels of 614,216 gross tons, according to the records kindly furnished in advance of his annual report by Mr. Eugene T. Chamberlain, the Commissioner of Navigation. This was a far greater output than the 1,157 vessels of 471,332 tons built and documented in the previous year. Of the 614,216 gross tons of shipping built in the fiscal year 1908, no less than 450,017 tons was composed of steel. To this great product, however, both in total construction and in steel construction, the Great Lakes of this country, doubly protected by geography and law, were the chief contributors. The steel steamers of over 1,000 gross tons built on the seaboard of the United States last year numbered 25, of 101,658 tons, while 58 steel steamers of 322,806 tons were launched by lake shipbuilding companies."

In 1910, just two years later, Commander Stevenson Taylor, in his address at the opening session of the meeting of the Society of that year, deplored the backward condition of our national maritime development, but with prophetic vision asserted that—

"Some day the people of this great nation will realize the supreme importance of having ships and yards in which to build them; will realize that we are deliberately passing to foreigners annually enormous sums which should be earned by our citizens; and they will come to themselves, will unite on the right course, will demand from their representatives a change from the present condition, and they will get it."

Little as Commander Taylor or any of those who heard or read his remarks dreamed of the changes which were destined to take place in a few years, his prophecy has come true, but in a manner and to a degree quite beyond the wildest expectation of any of those present at the meeting in 1910.

From the record performance of 80 steel vessels of about 168,000 gross tons referred to with such satisfaction by Mr. Griscom in 1910, and the record noted by Admiral Bowles in 1908 as the greatest in our history, when 1,457 vessels of 614,000 tons were produced in all the shipyards of the country, there is a giant stride forward in the developments which have taken place during the past three years, reaching their culmination in the year 1919.

This tremendous advance is perhaps shown most directly by the statement that whereas in the year 1908 the gross output of all vessels built in the United States was only 614,000 tons, of which only 450,000 tons was of steel construction, the output in steel vessels of more than 1,000 tons was in the year 1918 more than 850,000 gross tons, and for the fiscal year 1919, nearly 2,500,000 gross tons.

These figures are especially significant when we consider that the output for the year 1908 was the maximum production in any one year in the United States up to our entry in the Great War. Moreover, the greatest annual output in launchings of the world's shipyards prior to the year 1917 was 3,032,000 gross tons in the year 1913.

Expressed in another form, the increased capacity of our shipyards is even more impressive. The actual average monthly output of steel seagoing vessels for the first eight months of the year 1919 was nearly 300,000 gross tons, and the actual average monthly output for the last four months of this period was more than 350,000 gross tons. For the same period the average monthly total output of *seagoing* vessels was more than 400,000 tons, or at the rate of nearly 5,000,000 tons per annum.

Comparing these monthly averages with the world's pre-war five-year annual average of about 2,300,000 gross tons shows in another very explicit manner the tremendous recent expansion of shipbuilding facilities and output of ships in the United States.

The causes and purposes of this tremendous development are well known to you. It is quite unnecessary to enumerate or to dilate upon them. America, in its tremendous effort to do its share in the winning of the Great War, permitted no obstacle to interfere with the fulfilment of its promises. This extraordinary development in the shipbuilding industry has not been accomplished, however, without difficulty, and there has been temporary dislocation of other activities in our industrial world. The difficulties and the dislocation were, however, unavoidable, in large part, having in view the tremendous operations, of many and varied character, being conducted simultaneously. Results were the goals aimed at. Methods often were necessarily assigned a subordinate place. Nor has the signing of an armistice more than twelve months ago very greatly lessened the strain upon many of those who have been responsible for the vast undertakings in which the United States engaged during the World War. As a matter of fact the return to normal conditions can only be satisfactorily accomplished through the earnest effort and heartiest cooperation of all those directly concerned.

The condition so greatly desired several years ago, however, has arrived. As a nation, we have the facilities for the most extensive building of ships. As a nation, we have the most extraordinary opportunities for the operation of those ships. A consummation devoutly to be wished is that this capacity to produce and opportunity to operate ships may be so assisted by wise legislation that the employment of our shipping in meeting the many and complex problems of trans-oceanic transportation—so vitally important to the national interest and welfare—may be undertaken and continued under laws which will produce the best possible results.

The foregoing is only a brief and inadequate presentation of some salient facts concerning the expansion of a great national industry whose life is inextricably linked with the well-being of this Society. These facts, moreover, relate largely to material accomplishment.

While it is to be hoped that in producing quantity, quality has not seriously suffered, experience in such matters is our best guide. Tremendous expansion of personnel—artisan, managerial, designing—during the past few years has necessarily made severe demands on the small initial supply of such personnel. The expansion which has taken place has been more than sevenfold, and it would be quite too much to expect that in such a tremendous development the standard of quality could always be maintained.

It is believed that the years of readjustment now before us present great opportunities to those engaged in shipbuilding and ship operating.

In this period of readjustment of business and professional life, character of performance will be, in the final analysis, the measure of our success. As with the individual, so with professional and business life, high character ultimately compels success, and *quality* under normal conditions will be given inevitably the precedence to which it is entitled.

This Society from its inception has had for its object, as expressed in the second article of its Constitution, "the promotion of the art of shipbuilding, commercial and naval." Our Constitution also provides that—

"In furtherance of this object, annual meetings shall be held for the reading and discussion of appropriate papers and interchange of professional ideas, thus making it possible to combine the results of experience and research on the part of shipbuilders, marine engineers, naval officers, yachtsmen, and those skilled in producing the material from which ships are built and equipped."

The great and fundamental object of this Society, therefore, as expressed in its Constitution and as provided by the statutory laws under which it has been incorporated as a scientific society, make clear the path of our most helpful and healthful development.

While our membership embraces in its regular vocations a wide range of professional and business activities, those activities, so far as they relate to this Society, should serve primarily to promote the creation and diffusion of knowledge of the art of shipbuilding in its broadest sense and the development of the many arts and sciences related thereto.

The Society has, moreover, an especially important function in encouraging and developing, through the preparation and presentation of papers and otherwise, the latent genius and talent of its younger members, especially those whose attainments may not yet have qualified them for full membership.

In a sense, therefore, the Society is a great educational institution whose standards and ideals must be maintained, and whose powers of instruction and professional helpfulness should be constantly developed. To aim at less would be to miss our greatest opportunity for good.

Anything, therefore, which would seriously divert the Society as a whole from such purposes and ideals and tend, even in slight degree, to lower its standards is to be avoided in the future as in the past. For upon the maintenance of high standards "for the art of shipbuilding" depends in large measure the satisfactory solution of the many great problems connected with ocean, lake, and inland water transportation with their consequent beneficent results for humanity.

In view of the admirable collection of papers awaiting your consideration, I shall detain you no longer, but immediately proceed to the important business for which we have assembled.

THE PRESIDENT:—The secretary will now read the list of transfers to the various grades of membership.

THE SECRETARY:—The following applications of transfers of members from one grade to another of the Society have been received and favorably recommended by the Council:

Associate to Member (7)

Chas. T. Burton, Superintendent of Hull Construction, Standard Shipbuilding Corporation, Shooters Island, N. Y. P. O. address: 622 Henderson Avenue, West Brighton, N. Y.

Leroy E. Caverly, Assistant General Manager, Los Angeles Shipbuilding and Dry Dock Company, San Pedro, Cal.

Frank M. Hiatt, Charge Man, Bureau of Construction and Repair, Navy Department, Washington, D. C.

Fred T. Llewellyn, in Charge of Sales Office, U. S. Steel Corp., New York, N. Y.

Aaron Matheis, Draughtsman, Federal Shipbuilding Company, Kearney, N. J., Lieutenant, U. S. N. R. F. P. O. address: 71 Millington Avenue, Newark, N. J.

Bruce A. Russell, Assistant Chief Hull Draughtsman, Ames Shipbuilding and Dry Dock Company. P. O. address: Ames Shipbuilding and Dry Dock Company, Seattle, Wash.

C. L. Putzel, Ames Shipbuilding and Dry Dock Company, Seattle, Wash.

Junior to Associate (3).

Wm. Neal Briceland, Inspector, Morse Dry Dock and Repair Company, Brooklyn, N. Y. P. O. address: 328 73d Street, Brooklyn, N. Y.

M. L. Goldstein, Engineering Department, Newport News Shipbuilding and Dry Dock Company. P. O. address: P. O. Box 515, Newport News, Va.

Herbert L. Lilla, Chief Engineer, Mine Sweeper No. 43, U. S. S. Gebbe, Portsmouth, N. H.

Junior to Member (1).

H. C. Adams, Jr., 4232 Pine Street, Philadelphia, Pa.

The President:—You have heard the list read, gentlemen. What is your pleasure?

Professor Herbert C. Sadler, *Member of Council:*—I move that the list be adopted as read.

The motion was seconded, put to vote and carried.

The President:—The first paper on our program is entitled "Methods Employed in the Construction of Concrete Ships," by Mr. R. J. Wig, Visitor. In the absence of Mr. Wig, I will ask the secretary to present an abstract of the paper.

The Secretary:—I think, perhaps, rather than to attempt to read an abstract of the paper, which is being distributed, I will make a few remarks on the subject of concrete ships for Mr. Wig.

I would like to say, in the first place, that I had the pleasure of seeing Mr. Wig in action, so to speak, as he was one of the staff of the Emergency Fleet Corporation with which I was connected at the time. He had possibly the most difficult problem to attack that was presented to any person during the war who was trying to build ships of any kind, sort or description. There was a quite justifiable and very serious prejudice against the use of concrete for shipbuilding, unless it was an absolute necessity, and particularly until its practicability had been demonstrated. In other words, Mr. Wig, who was actually under instructions to design and get ready concrete ships that should be serviceable, was working against an enormous inertia, and further than that, he was attacking a problem concerning which there was little or no information, at least in the size and type of vessels which he was to design. I may say for him that he was, fortunately, an enthusiast. If he had not been, I do not think he could ever have accomplished what he did.

He collected around him a corps of men who were experts in concrete work and also a number of highly qualified engineers, and, in addition, men who really had been connected with shipbuilding before. He went into the matter most thoroughly. It was not only necessary to develop the art of concrete shipbuilding which had been begun, of course, before this, but to apply it to the much larger hulls and endeavor to make the hulls satisfactory when built.

It was essential that weight should be saved as much as possible without sacrificing the strength of the structure, and to that end very elaborate and exhaustive experiments were made to determine the best form of concrete to use in the hull work.

That Mr. Wig claims to have been satisfactorily developed, and I sincerely hope the results may prove that the concrete ships, of which there will soon be in operation a number of 7,500-ton vessels, will prove lasting and serviceable.

In order to make sure that his reinforced materials were properly disposed, so that he could get the necessary structural strength without relying upon the concrete, except as a form of rigidity, he endeavored to find, in so far as any record was available, what were the actual stresses, strains, deflections and distortions which had been experienced with other ships; in other words, he went at the problem as scientifically as he could—he wished to see what forces he would have to overcome, what stresses he would have to meet, and he did these things, and I hope succeeded in placing his reinforcing material so that he would get the best results. He really did use a great deal of determination, ingenuity and thought in that matter.

His previous paper, presented last year, gave a very clear idea of what he proposed to do, and the paper which is to be presented this morning is in amplification of the first. I am only sorry that Mr. Wig is not here to state his case, because he has a case to state. I am afraid that the majority of people here will argue against him. I am not advocating concrete ships. I think the matter is still to be proven, but the points of my remarks have been to try to express to the Society what an earnest effort Mr. Wig made in this direction, and also my conviction that he made a remarkable advance in the art of building concrete ships.

Whether the construction of vessels of this material is going to be matter that will be followed by subsequent construction or not is something we cannot tell, but, from observing these vessels built from Mr. Wig's design, we will certainly learn the best that could be accomplished in concrete construction up to the present time.

I sincerely hope that the members will read the paper carefully and will be interested in following the operation of these ships. For your information I may say that certain of them will be in operation in a very short time. The only concrete ship of considerable size that is known to the world at large, as being a ship that will float and operate, is the good ship Faith. I had the pleasure of inspecting that vessel after she had traveled some 12,000 miles, and at that time I saw her in the dry dock, and also floating. She showed little or no sign of deterioration, and I believe she has carried cargo many miles safely. I also have heard rumors that she is giving certain trouble. On the other hand, any experience with the Faith should not be taken as conclusive that concrete ships will not be serviceable. The Faith was hastily built, designed without any very careful investigation of the conditions she would have to meet, and I think Mr. Wig and his associates feel and know that their vessels are much better from an engineering point of view, and they should, therefore, have a much better chance to live and see a good many years of practical service.

METHOD OF CONSTRUCTION OF CONCRETE SHIPS.*

By R. J. Wig, Esq., Visitor.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

INTRODUCTION.

This paper is confined to a description of the methods employed in the construction of concrete ships for the United States Shipping Board, Emergency Fleet Corporation.

The complete program of the Fleet Corporation comprises the construction of fourteen ships of five different types. The characteristics of these ships are given in Table 1, Plate 1. Two of the above ships were considered of experimental character and constructed under contract in private yards. One of 3,000 tons deadweight was designed by the Liberty Shipbuilding Company, Wilmington, N. C. The hull was built by them in a private yard at Brunswick, Ga. The outfitting and installation of machinery were done under contract by the American Shipbuilding Company of Brunswick, Ga. The other ship was of 3,500 tons deadweight and was designed by the Fougner Concrete Shipbuilding Company of New York. The hull was constructed by the designers at Flushing Bay, N. Y., but the outfitting and installation of machinery were done by the Lord Construction Company of Providence, R. I.

The remaining twelve ships were constructed in yards† owned by the Emergency Fleet Corporation and especially designed and built for the purpose. The yards were built and operated by contractors who acted as agents for the corporation. Following is a list of the contractors and a statement of the location of the yards and the number and type of ships built.

The San Francisco Shipbuilding Company, operating the yard at Oakland, Cal., are constructing complete two 7,500-ton deadweight concrete oil tankers (Type 70) (Figs. 1a and 1b, Plate 5), and one 7,500-ton deadweight cargo ship (Type 69) (Figs. 2a and 2b, Plate 7). The Pacific Marine and Construction Company are operating the yard at San Diego, Cal. They are constructing complete two 7,500-ton deadweight concrete oil tankers (Type 70). Fred T. Ley & Co., Inc., of Boston, Mass., are operating the yard at Mobile, Ala., and are constructing complete two 7,500-ton deadweight concrete oil tankers (Type 70) and one 7,500-ton deadweight cargo ship (Type 69). The A. Bentley & Sons Company of Toledo, Ohio, are oper-

*It was necessary, on account of the cost of printing, to omit many illustrations furnished by the author, showing details of the construction work and the equipment used.

†For description of yards, see paper by A. L. Bush, "Layout and Equipment of the Government Concrete Shipyards," Pro. Am. Concrete Institute, 1919.

ating the yard at Jacksonville, Fla., and are constructing the concrete hulls for two 7,500-ton deadweight oil tankers (Type 70). The tankers are being outfitted by the Jacksonville Ship Outfitting Company at Jacksonville, Fla. The Liberty Shipbuilding Company are operating the yard at Wilmington, N. C., and are constructing two 3,500-ton deadweight concrete hulls (Type 2) (Fig. 3, Plate 6). The outfitting is being done by the Jacksonville Ship Outfitting Company.

The reinforced concrete ship is an ultra refined concrete structure with walls, floors and columns similar to a building, requiring the engineer to deal with forms, reinforcement and concrete, but with such refinement in quality and workmanship that new methods of construction had to be developed or old ones improved in order to meet the exacting conditions. Forms on a large scale had to be built of great rigidity to conform to irregular curves and shapes and exacting dimensions, and supported by novel means. Reinforcing steel, of both large and small diameter and long and short lengths, had to be bent with exactness to irregular curves and firmly secured in position with very small tolerances on account of limited spaces made necessary by the requirement for minimum weight of hull and small allowable covering of concrete. Concrete of excellent quality, but of the lightest possible weight, had to be placed in thin walls containing very high percentages of reinforcing steel, the bars being in many cases only a few diameters apart.

Unlike a building the concrete ship must be constructed on a temporary foundation or underpinning.

The Emergency Fleet Corporation adopted the policy of suggesting feasible methods to the contractors, but permitting each to develop and use its own methods provided the cost was not excessive. Thus the methods employed for the various operations differed in the several yards.

Complete cost records have been made of each operation, and comparisons can be made of the relative efficiency of the several methods.

Before discussing in detail the methods employed by the contractors for the various operations it would perhaps first be well to outline briefly the stages through which the construction work progresses, giving a general bird's-eye view of the entire operation. The order of procedure is not fixed, for many of the operations overlapped and it was not possible at times to proceed with the work as desired because of lack of certain required materials, but the outline here given was essentially followed.

1. The underpinning or blocking for supporting the floor forms is set in position on the ways.

2. The scaffolding with overhead trusses for holding the outside forms is set in position.

3. The outside bottom and side forms are erected complete, thus providing means for supporting the reinforcing steel.

4. All steel inserts such as sea chests, stern frame, stem plate, hawse pipes, etc., are secured in place on the inside of the outside forms.

5. The bottom and side shell reinforcing steel is placed within the outside forms.

6. The bottom and side frame steel and the keelson reinforcing steel is erected. Splice bars between the bulkheads and shell are placed in position.

7. The inside frame, keelson and side shell forms are erected to the height of 4 or 5 feet.

8. Concrete is placed in the keelsons and in the bottom and sides of shell and frames up to the 4 or 5-foot draught line.

9. Bottom inside forms are removed and the concrete is pointed up where necessary. Top surface of concrete, where it will join the succeeding pour of concrete, is thoroughly cleaned and roughened.

10. The erection of the frame and bulkhead steel is continued up to the elevation of the second deck.

11. The inside frame, bulkhead and shell forms are erected up to the second deck.

12. Inserts for pipes, equipment, etc., in frames and bulkheads are set in place.

13. Concrete is placed up to the under side of the second deck.

14. The inside forms are removed from the frames and bulkheads, the concrete is pointed, and the upper surface which makes a joint with the next lift is cleaned and roughened.

15. Forms for the second deck beams and slab are placed, supported on inside staging and props from the concrete keelsons and frames.

16. Inserts for equipment and pipes in the second deck are placed.

17. The reinforcing steel in the second deck is placed.

18. The concrete is placed in the slab and beams of the second deck.

19. The inside frame and bulkhead reinforcing steel is erected to the top deck.

20. The inside shell, bulkhead and frame forms are erected to the top deck.

21. Inserts for equipment and pipes in frames and bulkheads between the second and top decks are placed.

22. Concrete is placed in the shell, bulkheads and frames to the under side of the fillet at the top deck.

23. The upper surface of the concrete which makes a joint with the next pour is cleaned and roughened.

24. The top deck beams and slab forms are placed.

25. The deck inserts for equipment, pipes, etc., are placed.

26. The longitudinal reinforcing steel is placed in the fillet at the top deck and in the deck beams and slab.

27. Concrete is placed in the top deck fillet, the deck beams and the deck slab.

28. The reinforcing steel is placed in the hatch coamings, bulwarks and deck erections.

29. The forms are placed for hatch coamings, bulwarks and deck erections.

30. The concrete is placed in all deck erections.

31. All remaining inside forms and staging are removed and all interior surfaces of the concrete are cleaned and pointed. All outside forms on the sides and bottom of the hull are removed, and the concrete is cleaned and pointed where defective.

32. All tanks are tested up to the light draught line and pointing and patching of concrete is done if found necessary after testing.

33. The exterior and, if time permits before launching, the interior surfaces of the concrete are painted.

34. The launching ways are placed in position under the hull, the blocking is changed, and the hull is launched.

It will of course be impossible in this paper to discuss all the variations of methods which have been employed by each contractor for all operations. Consideration will be confined to the more important and general features.

BLOCKING, SCAFFOLDING, TRUSSES AND OUTSIDE FORMS.

All blocking, outside scaffolding, trusses and outside forms are erected complete before any concrete is poured, and in some cases before any reinforcing steel is erected.

The hull as constructed must be supported several feet above the building ways to make the bottom accessible for removal of forms, examination and painting of the outside of the hull and the installation of the launching ways. In steel ship construction it is common practice to use heavy timber in the form of blocking or cribbing for this purpose. There are three fixed stages of support required:—(1) Support during construction; (2) temporary blocking during the removal of bottom forms and painting; (3) the final blocking on sliding ways for launching.

In concrete ship construction, since a complete flooring or form supported on joists or stringers must be provided, it was possible, where ways were laid out for side launching, to use either a truss or block and crib type of support.

At the Wilmington and Jacksonville yards, the bottom form joists are supported at the proper elevation on bents of pre-fabricated trusses having upper and lower chords with vertical posts and diagonal bracing. At Oakland, a similar truss was used, but the floor form sheathing is attached directly to the top chord of the truss. These trusses are placed athwartships. The lower chord is supported upon wedges resting upon the pile caps of the building way, and the upper chord carries the form sheathing direct to the joists of the floor forms, which run longitudinally with the ship. At the forward and aft ends the form joists run athwartships, and longitudinal stringers are provided between trusses and form joists.

The trusses are divided at the keel into sections and taper from the bilges of the ship toward the keel to accommodate themselves to the sloping way and the dead rise of the vessel. Sway bracing is provided between trusses at regular intervals. It was necessary that the trusses be so spaced that alternate trusses could be removed to permit the removal of the forms, painting of the bottom, and the insertion of the temporary and final blocking. Heavy timber was used in all the yards in the form of blocking or cribbing for the second or temporary support during the removal of the forms and the painting of the hull, and also for the final blocking on the sliding ways.

The Mobile and San Diego yards used heavy timber in the form of blocking and cribbing at all stages of support. The details of blocking, however, are not the same in both yards. At San Diego girders athwartships are placed on the blocking and support the form joists which extend forward and aft. At Mobile, the girders are placed longitudinally and form the joists athwartships.

Blocking was used at San Diego for all stages of support. On the shipways transverse sills are arranged in groups that center under each alternate transverse hull frame to carry a system of blocking, gang wedges and girders which support the bottom forms during construction. Here the forms are built in sections consisting of removable panels of a uniform width and extend over two frame spaces where their fore and aft edges rest upon transverse girders. The girders for each section are supported on successive sets of blocking and are held to their proper elevation by adjusted gang wedges. At the forward and aft ends the bottom forms are not built in panels but as a unit.

Scaffolding is necessary to support the outside forms. The details of the design of the scaffolding differed somewhat in the several yards. The general scheme employed, however, was quite similar. Essentially the scaffolding consists of two lines of trestles built of light wood construction, which are anchored at the bottom to the sills of the building ways, and at the top, at regular intervals, they are tied and held apart across the ship by trusses of similar construction. In some cases these top trusses are placed quite close together (10 to 15 feet center to center), while in one yard only three trusses were used for the entire length of the ship (435 feet). The wide spacing of the trusses affords less interference with overhead handling of materials into the hull, and is to be preferred. Struts from the scaffolding held the outside forms in place.

The forms for ship construction must be smooth, very rigidly built and braced, and more exactly constructed than concrete forms for ordinary building construction. The concrete sections have been made a minimum in size which will accommodate the steel and meet the strength requirements, and any deviation in the form construction either results in added weight of hull, lack of space or cover for the reinforcement or lack of uniformity in the concrete surfaces.

The hull lines to which the outside forms had to be constructed were obtained from the mold loft floor by one of several methods. Either templets were made of the cross-section required for the midship body and every frame of the forward and aft body and the forms constructed in place from these templets, or the forms were prefabricated in panels, the lines being taken directly from the mold loft floor, or portions of the studding were cut to shape in the mold loft, thus giving the ship lines, when erected, to which the sheathing was directly applied. In most yards a combination of these methods was employed.

The thin board form of templet commonly used in steel ship construction was employed in all but one case where templets were used. At Wilmington, N. C., an adjustable templet was devised and successfully used.

This templet consists of a flexible batten attached to a rigid frame in such a manner that the batten can be bent to any shape desired, and rigidly supported in that position simply by tightening the bolts, which pass through strips attached at their ends to the flexible batten. With these "curved sets," as they were called, it was possible to lift lines from the mold loft floor and transfer them to the steel bending tables or to the form timbers. For further use the lines were transferred to heavy sheets of templet paper.

Cypress was largely used as outside form lumber in the east and Oregon pine in the west. Since the outside forms are exposed to the weather for several weeks or months before the concrete is poured, it is necessary that some material be used which will not warp badly. Most yards used light ⅞ or ¾-inch sheathing throughout. One yard used 1¾-inch sheathing in the bottom and bilge forms.

It was found to be impossible to secure the outside forms rigidly in position so that they would not spread and would withstand the vibration of the air hammers which were used in placing the concrete unless provided with a through tie to the inside forms. Another method is to tie both inside and outside forms to the reinforcing steel.

In one case the outside forms were prefabricated in panels throughout the entire ship. In all other cases the outside forms were prefabricated in panels only through the middle body portion of the ship and the curved portions, forward and aft, were built as a unit in place. This latter method has proven the more satisfactory.

At San Francisco and Wilmington the entire outside form was built as a unit in place, but joints were made in the bottom sheathing for removal in panels through the middle body section. A description of the method employed is given below. Its construction was comparatively simple. Shiplap, ¾-inch in thickness, was used horizontally, backed by 2 by 6-inch studs at the frames and midway between frames or on 25½-inch centers. The studs were fitted to the proper shape on the mold loft floor, allowance being made for the thickness of the shiplap. They were erected in the hull so that one face was coincident with the center line of the frame (Fig. 28, Plate 8).

For purposes of erection, wires were strung along the center line of the hull and also 27 feet on either side of the center line, 27 feet being the half breadth of the ship. The base line was also given at each frame. Points for stringing these wires were given by the surveyor.

While the ribs were on the mold loft floor, the 36-foot, 26-foot and 10-foot water-lines were scribed on them, and also the intersection of the bottom and sides located as in Fig. 28. The offsets at each water-line from both the center line and the 27-foot line were scribed on a batten at the mold loft, four battens being required. These battens were then used in setting the studs, the offset from the 27-foot side being used and the offset to the center line used as a check. A light batten *D* was fitted across the faces of the studs as in Fig. 28 to take the curve of the ship so that the distance *a* could be measured and set off on the stud as distance *c*. This, done at several places vertically on the studs, gave points on a line so that they could be adzed to the proper surface *DD* ready to receive the shiplap.

INSIDE FORMS.

With one exception all the contractors prefabricated the inside forms in sections ready to set into position without requiring any carpenter work to be done within the hull. The lines for the inside forms were obtained from the mold loft floor or from templets taken from the floor. These forms had to be constructed with great exactness, for in many cases there was only ⅜-inch clearance from the steel on all sides. To show the accuracy with which such forms can be made, it is of interest to know that in the case of one ship none of these prefabricated forms had to be removed for adjustment after being set in place.

There perhaps is no phase of the concrete ship construction work which is more complicated than the construction of the inside forms. A description of the exact method followed at one of the yards in obtaining the shape and dimensions of the various sections will be of interest.

The frame drawings gave the thickness of the shell and the depth and width of the frame, as in Fig. 30, Plate 9. The problem was to obtain the width of the frame forms *N, W,* and the panel form *P,* and also their relations to the mold loft line which passes through the point *ML* in Fig. 30 (*c*). The bevels across the frames at the critical places were first obtained. These critical places were at the bottom of the bilge *L,* top of the bilge *U,* and at the main deck. The method of obtaining the bevels was to measure the distance *a* in Fig. 30 (*a*) between half frames on the mold loft. This divided by the frame interval 51 inches gave the bevel across that particular frame between the two half frames. These bevels were taken in a plane perpendicular to the shell.

The next step was to obtain the dimensions *A, B,* and *C* of Fig. 30 (*c*) which were all functions of the frame bevels. As a great number of these had to be computed, graphical charts were made for convenience.

It will be noticed in Fig. 30 (*c*) that the diagonal distance *A* is given to the outside line of the shiplap form produced. This was done in order that a right-angled, beveled chamfer strip could be used at the intersection of the frame form and panel form as shown. The panel forms were so cut that their edges were in the same plane as the side of the frame.

Following is the form in which the information was given the mold loft carpenter:—

Frame		Lower bilge	Upper bilge	Main deck
14	A	6⅛	5¼	4⅞
	B	1¾	1⅛	⅞
	C	1 11	1 10¼	1 10

As the mold loft lines were straight, *i. e.,* not the conventional ship curves, forms for the sides *N* and *W* could be made up to approximate size beforehand. These would be brought in to the floor in pairs and laid down adjacent to the frame line ror which they were intended. It was then a simple matter to apply the dimension *A, B* and *C* at the critical points *M, U* and *L,* Fig. 30 (*a*), and obtain points which when connected gave the proper width of form for the entire length. At this stage all brackets and beams framing in to the frames as shown on the frame detail were provided for, so that the measuring and fitting in the ship were reduced to a minimum.

The next step was to assemble the *W* of one frame with an *N* of another and connect them with the panel forms *P,* Fig. 30 (*c*). Before the forms *N* and *W* were lifted from the floor, water-lines and offsets in even feet were scribed on; for example, in Fig. 31 (*a*), Plate 10, the 12-foot and 24-foot water-lines and 20-foot and 18-foot offsets. This, of course, coordinated them precisely in two dimensions and gave the proper position which they would occupy in the ship. There only remained to place them in some sort of a rigid rig which would maintain them in their proper relation, the correct distance apart, while the panel boards *P* were being nailed on and the whole form braced so that it could be handled and erected.

Fig. 31 (*b*), Plate 10, shows sketches of this rig. The panel boards were nailed to cleats on the sides of the forms in approximate length. They were then sawed with their edges in the same plane as the side of the frame, so that the right-angled, beveled chamfer strip could be nailed on.

It is worthy of note that the forms built up in this way fitted the varying bevels of the shell with great accuracy and no trouble was experienced in maintaining the proper shell thickness at all places.

INSERTS.

There are from 5,000 to 6,000 so-called "inserts" or fittings which must be cast into the hull of the concrete ship as the concrete is placed. These are composed largely of anchor bolts, sockets, pipe sleeves and miscellaneous holes which must be provided for attaching equipment and furnishing openings for pipes, drains, etc. There are also many large inserts which must be provided, such as stern frame, stem plate, hawse pipes, sea chests, outboard discharge fittings and the like. Large fittings like the cast-steel stern frame and stem shoe were anchored to the concrete by means of bars which were bolted or riveted to the fittings, and the free ends of the bars were cast into the concrete. There was a wide difference of opinion as to the type of anchorage which should be provided to secure the stern frame to the hull, and certain variations of the construction of this detail were permitted. Sea chests and all other large pipe connections were provided with large flanges on either face, and the reinforcing bars were placed between the flanges, which were later filled with concrete, the faces of the flanges being in a plane with the surface of the finished concrete wall.

Wherever possible, the inserts were placed on the forms immediately after the forms were erected and before the reinforcing steel was placed. Bolts, sockets, and similar fittings for securing equipment were usually set in fixed templets fastened in the forms.

It was found that many of the castings, such as bollards, winches, etc., were not drilled exactly according to plan, and many of the anchor bolts would not meet up with the holes in the castings. It was found that the concrete could be readily drilled and new bolts set, so this defect was not of great importance. It is preferable, however, to set the fittings on the forms with holding-down bolts in sleeves attached to the fittings, and pour the concrete around them, or another satisfactory method is to drill the base of fittings from the templets used for setting the bolts.

KIND AND QUALITY OF REINFORCING STEEL.

It was originally intended to use deformed square bars of the structural steel grade in sizes ranging from 1¼ to ⅜ inches. The square bar was selected with the belief that it could be concentrated more readily into a thin section than the round bar, and a deformed bar was specified on account of its bond value. The square bars were found impracticable because they assumed a twist or wind when bent to conform to the curves of the ship, thus requiring more space than the side dimension, and the deformed feature was found objectionable from the construction standpoint because it was very difficult to weave the deformed bars on account of added friction due to the deformations. The bond stresses in the ship members were generally low and therefore a deformed bar was not necessary, so it was abandoned in favor of the plain, round bar ranging in size from 1⅜ to ⅜ inches diameter, which has been used in all ships except the Atlantus and Polias, the two experimental ships built by the Liberty Shipbuilding Company and the Fougner Concrete Shipbuilding Company.

It was intended to use structural grade steel, but early in 1918 the War Industries Board ruled that, because of the great demand for this grade of steel (largely caused by the steel ship program), no more structural grade steel reinforcing could be rolled in the east. It was necessary to substitute rods rolled from the discard croppings from shell ingots. This material, which may be classed as a hard steel, had an ultimate tensile strength of about 95,000 pounds per square inch, and a yield point of about 60,000 pounds per square inch. To provide for the use of this harder steel, two alternate specifications were written. The first, for the structural grade steel, is that recommended by the American Society for Testing Materials, Standard Specification for Billet Steel, Concrete Reinforcement, Structural Grade. The second, for the shell discard material, may be epitomized as follows:—

1. The steel must be made from new billet, open hearth stock.
2. The steel must contain more than 0.06 per cent phosphorus.
3. Its physical properties must conform to the following minimum requirements:

Yield point	50,000 pounds per square inch.
Ultimate strength	80,000 pounds per square inch.

$$\text{Elongation in } 8''\% = \frac{1{,}200{,}000}{\text{Ultimate strength.}}$$

Test specimens must bend cold, without cracking on the outside of the bent portion as follows: Bars below ¾-inch in diameter must bend through 180 degrees around a pin whose thickness equals the diameter of the bar. Bars ¾-inch in diameter or above must bend through 180 degrees around a pin whose thickness is twice the diameter of the bar.

Typical results of tests of the steel are given in Table 2, Plate 2. This hard steel caused considerable difficulty in the large sizes, for it was very hard to bend to shape. Approximately 1,600 short tons of reinforcing steel were used in each of the 7,500-ton tank and cargo ships and 550 tons in the 3,500-ton cargo ship. The longest length obtainable was 60 feet.

HANDLING OF REINFORCING STEEL INTO HULL.

At all yards the shell and bottom and side frame steel was carried by hand labor from the steel yard or railroad car into the hull, through openings left in one side of the outside forms near the quarter points of the length of the ship. The frame and side shell steel was hoisted by hand or block and tackle from the top of outside forms. At one yard a temporary extension was made of the railroad track to the opening in the forms, and the steel was dragged by hand labor from the car platform directly into the forms.

After the bottom concrete was poured it was necessary to close the openings in the lower part of the outside forms, and the steel had to be hoisted over the top. At Oakland a power lumber hoist operating on the side of the staging was used to hoist the steel to a runway on the top deck, from which it was carried by hand on to the ship. At San Diego it was hoisted in bundles by tower revolving cranes and dropped into the ship. Mobile used a gantry crane, Jacksonville used derricks, and Wilmington used tower whirlers in the same manner.

FABRICATION OF REINFORCING STEEL.

At San Diego all reinforcing steel was sheared to exact lengths, and all bars, including unbent bars, were tagged according to detailed plans and sent to the forms ready to set into position, all joints and laps being fixed on the plans. In all other yards the unbent longitudinal and shell steel was sent to the forms without shearing or tagging, the staggering of joints and laps being provided for by instructions to the steel foreman. It is questionable whether the second method is equally as good as the first.

The bending of stirrups was originally done by hand methods in all yards. The bender consisted of an arm pivoted at one end and carrying a hardened roller which bent the stirrups around a fixed pin as commonly used in building work. At Jacksonville a unique power bender was devised and proved very satisfactory. It works in the same manner as the hand bender, except that an air cylinder constructed of standard pipe and fittings with leather cup piston is provided to operate the bending arm, thus eliminating one man as well as considerably increasing the speed of bending. This device is capable of bending two ⅝-inch stirrups at a time.

The heavy frame bars, ranging from 1 to 1⅜-inch diameter round, were bent by normal methods either on a hand-bending table or by a power-bending machine such as the McKenna. Curved bends were difficult to make cold, because the bars had a certain amount of spring and the templet or dogs around which the bending was done had to be set to allow for the recovery in the bar. The quality of bars was not uniform in this respect, which required testing each bar with the curved templet after bending. Only a fraction of an inch tolerance could be allowed for error in bending. At Jacksonville, Fla., all large bars with curved bends were bent hot upon a cast-iron bending table about a steel templet previously set, and where angle bends occurred in the same bars they were heated and all bends completed at the same time. Bars for hot bending were heated to a dull red in a forge built as a plate metal box about 3 feet wide, 3 feet high, and 60 feet long, filled with sand to about 12 inches of the top, at which level a perforated pipe with removable ⅛-inch pipe plugs was extended the entire length of the forge and supplied with compressed air at one end. A fire of coal and coke was built of any necessary length along the forge, and accelerated by removing the pipe plugs through the range of distance required to be bent and at as many places along the 60-foot bar as bends were required.

Hot bending proved very satisfactory as to accuracy, cost and speed, but considerable saving would have been made had two forges been provided for the one bending table. It was found impracticable to withdraw one heated bar from the fire and replace it with a cold bar to keep the heating process continuous without breaking up the fire and disturbing the heating of the other bars. The bending crew were not occupied in bending more than about one-third of the total time and could easily have bent the output of two forges whenever several bars were to be bent to one radius and resetting of the templets on the bending table was not necessary. That this process did not affect the strength of the bars is shown by the tests made at the Bureau of Standards (Table 3, Plate 2).

The vertical shell steel usually ⅝ or ¾ inch diameter round, which has to conform where it is set into the bilge to the curvature of the ship, was bent on a hand-bending table if the curve was of large radius, or in power machine if of small radius or angle bends.

Where this relatively light steel of long lengths was bent at a distance from the point of construction, difficulty was experienced in some of the yards in transporting it to the forms and maintaining the original bend. At one yard the bending table was placed within the hull forms, and the vertical shell steel was carried directly from the bending table and set into position in the forms. If the other work to be done in the hull has been carefully planned, so that the steel bending and erection gangs will not interfere with other labor units, this method is quite satisfactory. It is preferable to have all possible work done outside the hull forms, for only a limited number of men can work efficiently within the hull.

PLACING OF REINFORCING STEEL.

The exact method of placing the steel varied, depending partly upon the method employed in supporting the steel. Following is a description of the method employed at San Diego for placing the shell steel.

On the inside of the outside forms the location of the outboard longitudinal bars was marked off, starting at the keel. On these lines nails were partly driven into the forms in transverse rows about 8-foot centers, indicating the location of the bars. The outboard steel is then placed directly in position on these nails as it is carried from the car.

It was more convenient for the workman walking back and forth if all the side shell steel is placed before placing the bottom steel. After placing the side, horizontal, outboard steel, the vertical side shell bars are placed, followed by the diagonal spacing bars and the horizontal inboard shell steel. The side shell steel is held away from the forms by means of small cement blocks wired to an occasional bar of the outboard layer of steel. The floor or bottom steel is then placed in a manner similar to the side shell steel, except that cylindrical cement blocks about 2 inches in diameter and of varying heights are used to support the steel off of the forms. A cement block, 2 inches high, carrying the second or transverse layer of reinforcing steel, was found to be preferable to a 1-inch block carrying the outboard steel, as the latter would occasionally crush. Where the higher block was used the outboard layer of longitudinal shell steel was suspended to the transverse bars with No. 16 gage annealed wire ties attached to every alternate transverse bar.

Staging was erected on the inside of outside forms for the use of workmen in handling the side shell and frame steel. Three different methods were employed by the several contractors. The method of hanging the staging from the top is preferable in that it keeps a clear space for handling the steel at all elevations along the sides of the forms and requires no holes through the sheathing.

San Francisco used ⅜ by 1¾-inch wood strips placed transversely at each frame, lightly nailed to the inside of the outside forms for supporting the shell steel off the forms. Jacksonville used ⅜-inch round bars in the same manner as the wood strips above, but only across the bottom forms. The wood strips were removed after the concrete was placed and the surface was pointed up. The round rods were not removed, however. For supporting the side shell steel, cubes of concrete were made with a groove in the top. These cubes have a hole through the center perpendicular to the groove for bolting to the inside of the outside forms, and the groove is so placed that the horizontal rod will be supported ⅜ inch from the forms. The nut of the bolt is placed on the inside so that after casting the concrete the bolt can be removed and the hole filled with cement. The cubes were spaced about 6 feet longitudinally and 3 feet vertically. The horizontal rods between the cubes were hung to the rods on the cubes by S-shaped wire hangers. As the vertical shell steel was placed the horizontal bars were wired to it and thus securely held in position. At Mobile, small cement or ceramic tile blocks were used to support the steel from the forms, and a special prekinked wire was used to space and support the shell bars. The most satisfactory method was probably the one used at San Diego, as described above. At Mobile, the shell steel was supported by the "White Bar Clip." The horizontal rods are hung from wires nailed to the forms and held in place by the clip.

Various means were employed for erecting the frame steel. Some form of clamp was used in three of the yards. In some cases where the clamps were used the outboard frame bars were individually wired into position, and the inboard frame bars were preassembled in the clamp and placed in position as a unit. At San Francisco the side frame bars were erected and then the bottom frame bars. In one yard a temporary wooden horse was placed at the elevation of the bottom of the vertical side frame bars until they were wired into place with the stirrups. The horse was then removed, and the bottom sections of frame bars were placed in a similar manner. At Wilmington, N. C., the frames were prefabricated in three sections. The first unit consisted of the bottom or floor section up to the 6 foot 4 inch water-line, and the other two sections extended from this line up to the deck line and over to the hatch girders or the center line of the ship. The prefabrication was done in an adjustable wooden frame, which was set to a templet for each frame. No difficulty was experienced in setting these prefabricated frame sections. The bottom sections of the frames were picked up by their ends, which tended to shorten them slightly, due to the sag, thus providing plenty of clearance to enter the forms. As soon as the frame was set in the floor on position and released, it sprang back into position, tight up against the shell steel. It was anticipated that there would be difficulty in setting the upper sections of the frames because of the lapping of the end bars of the two sections. Little difficulty, however, was experienced on this account.

Many special tools were designed to assist in erecting the steel.

The deck steel was placed in a manner similar to placing steel in floors in ordinary building construction. At Wilmington the large longitudinal bars in the fillet of the deck were electrically butt welded by means of a type 10-A 220V, 60 cycle, AC machine manufactured by the Federal Machine and Welder Company, of Warren, Ohio. This machine is equipped for the continuous welding of bars from ½ inch to 1¼ inches in size, either round or square. The bars come in lengths of 60 feet, and the butt-welding machine was set on the deck about 60 feet from the aft end of the hull. The bars were passed through the machine and over a series of iron rollers extending along the fillet. No difficulty was encountered in handling the bars or in placing them. Eight men could handle a 240-foot 1⅛-inch bar and place it in position in the fillet. Good welds were obtained in from 30 to 60 seconds. The hard steel was a little more difficult to weld than structural grade, which was not obtainable at the time this work was done.

CLEANING OF FORMS FOR CONCRETING.

The cleaning of the forms preparatory to placing concrete is one of the very important matters in concrete ship construction. On account of the relatively large amount of reinforcing steel in the structural members it is very difficult to remove sawdust, shavings and other debris which may collect at points of intersection of the steel such as where the keelsons, frames and bulkheads join the floor slab, and in the haunches in frames. If such debris has been permitted to accumulate at these points, both water and air, even under considerable pressure, have been found to be ineffective for its removal. It is practically impossible by visual inspection to detect such accumulations after the steel is erected.

Large openings should be left in the outside forms under all keelsons, bulkheads and the like, and at frequent intervals elsewhere over the bottom forms, and constant vigilance should be maintained to keep the forms clean during the erection of the steel and inside forms. No unnecessary carpenter work should be done within the hull, and such wood-working as is done should be over a work-box so that the waste will not fall into the forms. It is particularly difficult to see the accumulation of debris under such masses of steel at the frames, bulkheads, etc. The form was washed and supposedly cleaned, and an inspection before pouring did not detect the dirt.

All forms were thoroughly wetted just before placing concrete, and in some cases it was necessary to spray the reinforcing steel in order to reduce its temperature from exposure to the sun.

MATERIALS USED FOR CONCRETE.

A concrete was required which would have a compressive strength of not less than 4,000 pounds per square inch at 28 days and be of the lightest weight possible. While in the plastic state, it must be of such a consistency that it is possible to work it into place, thoroughly imbedding the reinforcing steel and completely filling the forms.

In order to obtain the highest possible strength with a maximum quantity of aggregate, the specifications called for a special high-grade Portland cement, which required that it be ground so that at least 90 per cent passed the 200-mesh sieve. The value of this fine grinding is shown by the results of tests given in Table 4, Plate 3.

It was necessary to develop some new type of aggregate in order to reduce appreciably the unit weight of the concrete. After much investigation an aggregate was developed with which it was possible to decrease the weight of the concrete from 145 pounds per cubic foot to 105 to 120 pounds per cubic foot. Since there are 2,800 cubic yards of concrete in each of the 7,500-ton ships, each pound reduction in the unit weight per cubic foot of the concrete represented an added carrying capacity of approximately 32.5 long tons in the ship. Thus a saving of 30 pounds per cubic foot represents approximately 1,000 long tons additional carrying capacity in the ship.

The aggregate developed is a vesicular slag* made by burning suitable clay in

*See *Engineering News-Record*, April 24, 1919, for complete description of product and method of manufacture.

either brick or cement kilns to such a temperature that it bloats. After cooling it is crushed, screened and used as any normal aggregate. The fines passing a 1/10 or 3/16-inch screen are used as sand, and the coarse from 1/10 or 3/16 to ½ inch is used as coarse aggregate.

This aggregate, being porous and very sharp, was not as workable in the plastic concrete as might have been desired, particularly if the fines lacked in content of dust. The gradation of the fines varied, depending in part upon the type of crushing equipment employed.

In order to increase the workability of the plastic concrete in such cases, a very small quantity (1½ per cent by weight or about 12 per cent by volume of cement) of celite was employed. This material is a very light-weight (about 12 pounds to the cubic foot) natural silica product of the following composition:—

Silica (SiO_2)	86.00*
Ferric oxide (Fe_2O_3)	1.61
Alumina (Al_2O_3)	2.99
Lime (CaO)	.44
Magnesia (MgO)	.72
Sulphuric anhydride (SO_3)	Trace.
Ignition loss	7.90
Alkali by difference	.34
	100.00

The addition of this material also tended to prevent segregation of the coarser particles in the concrete.

The concrete mixture employed was usually composed of one part by volume of cement to two parts of total aggregate. The proportion of coarse and fine in the aggregate varied from ⅔ part fine to 1⅓ part coarse to 1 part fine to 1 part coarse. Table 5, Plate 4, gives a summary of some of the tests made from concrete used in the hulls.

EQUIPMENT USED TO TRANSPORT CONCRETE TO FORMS.

A different type of concreting plant was used at each yard. At Oakland the plant consisted of two 24-cubic feet Koehring mixers set in pits. The aggregate and cement was brought to the mixers in buggies. A 90-foot elevator tower with one cubic yard automatic dumping buckets was set over each mixer. The buckets dumped into a hopper at the top of the tower. One double drum electrically operated hoist served both elevators. The forward tower distributed concrete through three main chutes, which terminated in one yard, distributing hoppers on the center line of the ship about 40 feet apart. The after tower supplied three similarly located distributing hoppers in the after end. These distributing hoppers were arranged with four outlets, and from them the concrete was passed through chutes to buggies and distributed by the buggies to the forms.

*16.76 is soluble in 10 per cent HCl and 11.24 is soluble in 5 per cent Na_2CO_3.

Since the hulls in this yard had been completely formed inside to the second deck before any concrete was poured, a system of signals had to be installed in order to keep the men in the various compartments supplied with concrete as needed. For supporting the runway upon which the concrete for the top deck was transported in buggies a hanging scaffold was employed.

At Wilmington and Jacksonville the concrete was transported from the mixers by bottom dump buckets swung on a whirler or derrick to a number of hoppers mounted on the staging at the top of the forms. The concrete was conducted by gravity through pipes to mortar boxes, from which it was shoveled or "pailed" to the forms. At Mobile the concrete was elevated in a bottom dump bucket by a gantry crane and swung directly to a mortar box, where it was deposited. From the mortar box the concrete was placed in the forms by shovels and pails.

The most unique equipment was that employed at San Diego. No. 24 motor-driven Koehring mixers located in pits discharge into Inslee controllable one-yard bottom dump buckets set on push cars that are transferred from under the discharge chutes of the mixer to a position on the arc of a circle prescribed by the end of a revolving tower crane boom at a fixed slope. From this position the bucket is elevated and swung into position over the hull, lowered between the trusses, where the concrete is dumped into a hopper which travels along a runway on the center line of the ship. The concrete is then discharged from the hopper at any point along the hull into chutes and delivered directly into forms or into flat boxes, and then handled by shovels and coal buckets into the forms. There are four traveling hoppers, operated on wooden tracks secured above the runway, which is hung from the trusses.

PLACING CONCRETE.

On account of the large amount of reinforcing steel and small clearances between the forms and steel it was found that the concrete could not be placed by the ordinary method of rodding or tamping unless a very watery concrete mixture of low strength was employed. As can be seen from an examination of the photographs, much of the interior of frames, keelsons and shell could not be reached with a rod. The most thorough and practical means of settling the concrete into the forms and about the reinforcing steel was found to be by vibration. After some experimentation it was found that small air hammers of commercial type (similar to Ingersoll-Rand "Little David" No. D), with blunt bitts from 12 to 36 inches in length, held against the outside or inside of the forms, near the point where the concrete was being placed, would not only cause the concrete to flow, fill the forms and thoroughly embed the steel, but would also increase the density of the concrete by driving out much of the entrapped air. Care was exercised not to let the hammer come in contact with the reinforcing steel. From thirty to sixty of these hammers were employed on each ship during the concreting. For horizontal surfaces such as decks, a long shank bitt was placed in the hammer, and in most cases applied to the upper side of the forms through the concrete.

It was at first attempted to vibrate the outside forms by maintaining a hammer crew outside of the hull, but this method was found unsatisfactory, for the hammers would at times through error be applied to surfaces where the concrete had been placed and partially set, or they would not be applied where they could most effectively settle the concrete. It was found more satisfactory to use a long shank bitt and apply the hammer from the inside directly to the floor through the concrete as it was being placed, and to the inside surface of the outside shell forms through the windows left in the inside forms, or through small holes made in the inside forms, which were later plugged.

The ordinary type of hammer, while suitable for vibration of the forms, requires the operator to absorb the recoil of the blow, and in consequence it is necessary to relieve him at frequent intervals. At San Diego a special type of hammer was devised, with an envelope casing which almost entirely absorbed the recoil. This hammer has proven to be very satisfactory.

There is a total of approximately 2,800 cubic yards of concrete to be placed in the 7,500-ton ships, distributed approximately as follows:—600 cubic yards in bottom to top of bilge, 1,200 yards from top of bilge to and including the second deck, 800 yards from second deck to and including main deck, and 200 yards in superstructures. These various quantities were placed in one continuous operation at the rate of from 8 to 15 cubic yards per hour.

With the large yardage of concrete to be placed in a continuous operation over a relatively large area in small units, it was necessary to organize the crews with considerable care and to plan a systematic operation. In some of the ships the placing of concrete was confined to two groups, either starting at opposite ends of the ship and working toward the middle or starting at the middle of the ship and working toward the ends. In most ships the depositing of concrete was carried on simultaneously in four sections of the hull; either two groups worked from the ends toward the middle and another two worked from the middle toward the ends, or two groups each started from the third points in the length of the ship and worked simultaneously toward the middle and ends of the ship.

There is a difference of opinion among the contractors as to whether the concrete should be placed in the bottom floor and allowed to flow under the bottom frame forms before filling the frame forms, or whether the bottom frame forms should be first filled, the concrete being permitted to flow out underneath into the floor before filling the floor forms. Defects have been found in some of the concrete placed by both methods. The results obtained are apparently dependent upon the care exercised.

Since the concrete was deposited continuously over periods of from fifty to one hundred hours, the men worked in two or three shifts of eight to twelve hours each. Great care and attention must be given to the placing of each shovelful of concrete into the forms, or defects of a serious nature will result. All the workmen connected with the placing of concrete should be schooled and instructed with regard to their individual responsibilities, for inspection alone cannot insure a good job, and one careless workman in this position can cause major defects.

No difficulty was experienced in making watertight construction joints between the several pours of concrete. The surface where new concrete was to be joined to old was thoroughly roughened by chipping with the pneumatic hammers, and any soft film of cement which might have formed was removed. The surface was then cleaned with compressed air or water and thoroughly saturated with water just before the placing of concrete of the next lift was started. In some cases the surface of the hardened concrete at the joint was grouted also, but this was not necessary with the very rich mixture employed, if care were exercised to insure that the first batches of concrete were thoroughly worked into the surface.

PATCHING AND POINTING THE CONCRETE.

Defects of at least a minor character have been found in the concrete of all hulls. This has necessitated some patching and pointing.

In most cases the patching was done by hand. The cavities were well cleaned, all loose material and dust removed, and the old concrete thoroughly saturated with water. A mortar, usually of one part cement to two parts of sand, was mixed with water to a moist earthy consistency; that is, it was just wet enough so that when formed into a ball in the hands it could be made to cling together. This mixture was then pounded into the cavity with a hand hammer and hard wood stick or block. Excellent results were obtained by this method, and practically no leaks have occurred around patches.

In some cases, holes were found which extended entirely through the shell, or the defects extended to such a depth that it was necessary to cut away the concrete through the shell. In these cases the repairs were made either by placing forms on both sides, securing them into position by bolting through, and placing the concrete as originally, or by placing a form on one side only and depositing the concrete as described for patching.

Where care has been exercised, the patching has been uniformly satisfactory and watertight walls have been secured.

The inspection must be very thorough in order to uncover all defects, for in some cases the surface may appear sound and uniform or show only a slight cavity, but by tapping it will be found that the cavity extends for some distance into the mass.

The cement gun has been used successfully in some cases for patching, but satisfactory results are obtained by the hand methods if carefully done.

LAUNCHING.

All of the concrete ships described in this paper, with the exception of the two small ships built in private yards, were built for sideways launching.*

*The reasons which led to the adoption of side launching are outlined in a paper by A. L. Bush, "Layout and Equipment of the Government Concrete Shipyards." Pro. Am. Concrete Institute, 1919.

The transferring of the hull from the ways to the water is accomplished in the following manner.

Alternate sections of the blocking and forms which supported the hull during construction are removed, and a temporary cribbing is inserted and brought to bear by driving in wedges. As the forms and original blocking are moved, the concrete is pointed, patched where necessary, and painted.

The remaining sections of form blocking are removed, and the patching and pointing of concrete is completed. The launching ways are set in place. The launching packing is then placed, the temporary blocking removed, transferring the load to the ways through the packing, and the ship is launched.

It is obvious that the details of the first two operations will vary with the method used to support the bottom forms. As was outlined in the section dealing with blocking, scaffolding, etc., the forms at San Francisco were supported on prefabricated trusses running transversely across the ship. These trusses were spliced at the center line and tapered from the bilges down to the keel. After concreting, the proper lines of trusses were pulled out from under the ship, allowing the forms to be removed. A similar method was used at Jacksonville and Wilmington.

At San Diego blocking was used instead of the trusses. Two of every three panels (each panel is 8 feet 6 inches) were removed and the load carried on the third. The exposed surface was painted, cribbing installed, and finally the last section of forms removed.

At Mobile a combination of blocking and struts was used to carry the bottom forms. Immediately before launching the weight of the ship is carried to the foundations through the temporary blocking and through a series of wedge blocks placed under the keel. The launching packing is in place on the greased ways, but does not, as yet, carry any load. The first operation of launching is to drive in wedges set in the packing, so that the packing will bear against the ship and thus take some of the hull's weight. This may be done in one operation, but the general practice is to set the wedges up in two or more rallies. The temporary wedges and keel blocks are then driven out with battering rams, leaving the entire load on the packing. The blocking is usually made up with a block wedge, held together by a steel strap and pin, and with its sloping faces heavily greased. When the pin is driven out with a sledge the block may usually be knocked apart with the ram. Provision is always made, however, to split it out with steel wedges in case it jams.

With the blocking out, the ship is restrained from sliding by a series of so-called "daggers" and "triggers" at the bow and stern. The thrust of the ship is taken into five daggers, each of which bears on a trigger. The thrust is carried across the triggers chiefly to a bolster block bolted to the side of the way and thoroughly braced by either one or more spins. The dagger is set so that the produced line of the face nearer the way clears the bolster by about an inch. The trigger is prevented from kicking out by a heavy line which passes under the ship and is anchored to a deadman on the inboard side of the building way. Each of these slings is carried over a chopping block, at which a man is stationed with a broad axe. After all blocking has been removed, the launching master, standing amidships on the inboard side, gives a signal, and all the lines are cut. The trigger is kicked out by the dagger which falls clear, and the hull slides down the ways.

The system just described was the one used at Mobile. Under the remainder of the ship there were thirty-five launching ways 8 feet and 6 inches on centers, each coming under a frame. The packing was built up to extend across three ways, and there were four sets to a transverse section of the ship, exclusive of the keel packing, which is individual for each way. These sets do not line up transversely, but each successive one starts one way further astern, so that one transverse set of packing rests on six different ways. For example, numbering the ways back from the bow, the first complete outboard packing rests on ways 7, 8 and 9. The set just inside it is on ways 8, 9 and 10; the one inside that on 9, 10 and 11; and the inboard set on 10, 11 and 12. This staggered arrangement was used so that there would be a greater certainty of all sets of packing starting to move at the same time.

At San Francisco the details differed materially from those used at Mobile. Instead of the conventional dagger and trigger release, they used a cradle both fore and aft. After the load had been taken on the packing, lashings from these cradles to deadmen restrained the tendency of the hull to slide. It is also to be noted that the packing extends across two ways instead of three, as was the case at Mobile.

This packing was built up in five transverse sets, one under the center keelson, two under the bilges, and two under the longitudinal bulkheads. The sets were not staggered, but two pieces of 2 by 8 ran transversely, connecting the five units of each set.

At the date of writing no 7,500-ton tankers have been launched at any yards other than Mobile and Oakland. At Jacksonville and San Diego, the method used at Mobile, with possible slight variations, will be followed.

The chief difference in the two methods described is in the releasing device. Mobile, with the dagger-trigger method, followed very closely the conventional side-launching procedure used almost exclusively on the Great Lakes and to some extent along the Atlantic seaboard. This method is just as positive and probably a little safer than the cradle scheme used at Oakland. In the latter it is necessary to use heavy timbers, in fairly long lengths, rigidly fastened together. Experiments conducted by the Concrete Ship Section on the sideways launching of steel ships show that a velocity of 23 feet a second is attained. It is obvious that if any of the cradle timbers should work loose and jam that the force of a 5,000-ton hull moving at this speed would be more than ample to punch the timber through the shell. In the trigger arrangement no such long sticks are present. This actually happened on the side-launching of a concrete barge when a cradle timber broke, jammed in the mud just forward of the launching ways, and ruptured the shell of the vessel.

Some of the dimensions and figures of the Mobile launching system follow:—

Slope of launching ways, 1¼ inch to the foot.
Slope of trigger ways, 1¼ inch to the foot.
5 triggers forward and five aft.
35 launching ways. All ways spaced 8 feet 6 inches on centers.
Launching ways, 18 inches by 30 inches hard pine.
Packing (mostly), 12 inches by 12 inches hard pine.
Bearing stress on grease, 2¼ tons per square foot.
Trigger rope, 1 inch diameter Manila hemp.
Launching grease:
  1/16-inch stearine on slides and ground ways.
  3/16-inch launching grease on slides and ground ways.
  ⅜-inch launching grease between packing and ways.

At the date of writing, the Fleet Corporation has launched three 7,500 D. W. T. concrete tankers and one 3,500 D. W. T. cargo ship by the side-launching method described. In addition, one each experimental 3,000 and 3,500 D. W. T. cargo ships have been launched endways. All six launchings have been thoroughly successful and without accident or damage to the hulls.

PAINTING.

Both the exterior and interior of all ships are painted.

The disintegration of reinforced concrete in sea water is due to the penetration of the water into the pores of the concrete in the submerged portion of the concrete, its absorption by capillarity into the air-exposed portion and the evaporation of the water with a resulting concentration of sea salts, causing an accelerated corrosion of the embedded reinforcement which splits and spalls the concrete.

The effectiveness of the protection to the steel provided by the rich concrete mixture used in ship construction is unknown.

As a precautionary measure, it was deemed wise to apply a paint coating to the exterior below the water-line to reduce penetration, and above the water-line and on the interior surfaces to reduce evaporation of any water which may be absorbed by capillary action. In the tank ships it was also deemed advisable to apply an impermeable semi-elastic oil-proof coating to the inside surface of all oil compartments. The bottom of a concrete ship will foul to the same extent as a steel ship unless it is protected; therefore an antifouling paint was also applied.

Many tests have been made and a large number of different paint combinations have been tried, but experience has not been sufficiently extensive to fully demonstrate their value. It would appear that the bituminous paints are the more satisfactory where continuously exposed to water. Following is a brief description of the various combinations now being used on the different surfaces.

All paints except the bituminous coatings were applied by air brushes.

It was originally intended to apply two priming coats of a 7½ per cent solution of magnesium fluosilicate over all surfaces of the hull. The purpose of these coats was to neutralize the alkali and harden the surface. This treatment was necessary where it was intended to apply an oil or varnish paint. It was found, however, that the varnish paints were of uncertain durability on the exterior surface of the hull, and a bituminous paint was substituted which obviated the necessity for the application of the magnesium fluosilicate solution. In the case of one hull a rapid oxidizing vegetable oil known as "Repello" was substituted for the magnesium fluosilicate. This material not only neutralizes the alkali but considerably reduces the absorption of the concrete.

The following combination of coatings has been applied to the outside surface below the light draught line—either two coats of magnesium fluosilicate, two coats of spar varnish and one coat of anti-fouling paint or three coats of bituminous paint and one coat of McInnes anti-fouling paint.

The outside surface above the light draught line is covered with the same coatings as the surface below the light draught line except that the anti-fouling coat is omitted, and where the two coats of spar varnish have been applied it is followed by a third coat of grey enamel.

The weather deck is covered with an asphalt mastic ¼ to ⅜ inch thick, which acts as a wearing surface and will prevent leakage through cracks if any occur.

The interior surfaces of all oil compartments are covered with two coats of magnesium fluosilicate and three coats of spar varnish. In some cases a grey enamel is substituted for the third coat of varnish. Cheese cloth is placed as reinforcement between the layers of varnish on the oil-exposed surface of all bulkheads which are exposed to oil on one side only. Cheese cloth is also placed in a similar manner on the inside surface of the shell of the hull between frames in the oil compartments.

The interior of all water tanks are covered with either three or four coats of bituminous paint.

The interior of cargo holds and boiler and engine-room compartments are coated with either one or two coats of magnesium fluosilicate and three coats of spar varnish; or two coats of magnesium fluosilicate, two coats of spar varnish, and one coat of grey enamel; or three coats of bitumen.

OUTFITTING.

After the concrete hull has been launched, the task of installing the machinery and piping, erecting the cargo-handling apparatus and rigging, and fitting up the accommodations for the officers and crew still remains before the ship is ready for sea. These items are all included under the heading of "Outfitting."

At the yards in Mobile, Oakland and San Diego, the superintendent who constructed the hull for the corporation is also doing the outfitting, and from these yards the ships are to be turned over ready for service. The ships built at Wilmington and Jacksonvile are surrendered after launching to the Jacksonville Ship Outfitting yard at Jacksonville, Fla.

The first of the three items mentioned, the installation of machinery, is the most important part of the outfitting. Besides the main engine and boilers, it includes all the auxiliaries such as pumps, condensers, stearing gear, refrigerating machinery, generators, capstans, windlass, and cargo winches.

In the 7,500 D. W. T. ships, both cargo and tankers, the propelling equipment is the same. The engine is triple-expansion, three-cylinder, vertical inverted direct-acting Stephenson link type, with cylinders 24½ inches, 41½ inches, and 72 inches in diameter with a 48-inch stroke, and is to develop 2,800 horse-power at 88 revolutions per minute (corresponding to a cross-head speed of 700 feet per minute). It is to operate at this speed with 200 pounds steam pressure. Steam for this engine is generated in a battery of three oil-burning Foster water-tube boilers, which are located in a boiler-room just forward of the engine-room. Each has an external heating surface of 3,050 square feet, and is to be built for a working pressure of 225 pounds per square inch.

Both engine and boilers and all the rest of the outfitting equipment are practically the same as those used on a steel ship of the same size. The only points of difference are in the details used to fasten the various pieces of equipment to the hull. The heavier machinery and the boilers are supported on steel grillage, which is fastened to the frames by means of bolts set in the concrete.

At Jacksonville the steel grillage was omitted under the engine, and a concrete foundation was provided to which the engine bed was directly secured by bolts passing through pipe sleeves cast into the concrete.

The location of these inserts is determined by means of templets on which the bolt holes are spotted either from the machinery drawings, or, better still, from the base of the machinery itself. For some of the other lighter members such as ladders, etc., the pipe sleeves are sometimes omitted, and a bolt with a large washer is placed directly in the concrete. The deck machinery subject to much vibration, such as cargo winches, windlass and capstan, is set on 2-inch oak bolsters. In the case of some of the smaller equipment, such as mooring bitts, the part is placed in position on the forms before concrete is poured.

It has been found difficult to place the smaller bolts and inserts so that they would fit the equipment unless they are set with templets taken directly from the casting. The drilling of ship castings and fittings is usually not exact according to drawings. Very little difficulty has been caused on this score, however, for if a bolt does not line up with the hole in the fitting a new hole is easily drilled in the concrete and a new bolt grouted in place.

The cargo-handling equipment of the 7,500-ton cargo ships is similar to that on a steel ship of the same size. There are two masts (see Fig. 3, Plate 7), each with four 5-ton cargo booms and each serving two hatches. In addition, one 30-ton boom can be rigged to the forward mast if necessary. When this is used it is necessary to shore the decks under the boom and to install additional guys, as the regular shrouds will not carry the load.

On the tankers no such elaborate equipment is necessary. Here most of the cargo is handled by pumps located in special pump-rooms adjacent to the main engine-room. The installation of the oil piping is similar to the installation in a steel ship of similar type. In Fig. 2, Plate 5, the holds for the oil cargo are marked "Cargo Oil." Immediately forward and aft of these compartments are small holds for dry cargo. To serve these holds each of the two masts is provided with one 5-ton boom.

There is very little to be said in regard to the fitting out of the crews' quarters, for they differ in no way from the quarters on any of the other Emergency Fleet ships. The exterior walls of the deck erections on all the ships are of concrete except the poops on the 7,500 D. W. T. ships, which are of wood bolted to the concrete by through bolts. None of the crew are quartered in the forecastle. The eight seamen have two large staterooms in the poop, and the rest of the ship's company are quartered in the bridge-house.

TRIAL TRIPS.

After the outfitting and installation of machinery has been completed, it is customary to subject the ship to two tests before it is accepted as ready for service. The first test is made while the ship is tied at the wharf and is known as the "Dock Trial." At this time the boilers are fired and the engines run at full speed for a time long enough to satisfy the trial board as to their proper performance, and the auxiliary engines are tested.

After the dock test the defects discovered are remedied and the second and final test made. In this test the ship is taken to sea and made to perform all the evolutions to be encountered in actual service.

At the time of writing, only the 3,000 and 3,500-ton D. W. experimental concrete ships have made their trial trips.

On the trial trip of the Polias (3,500-ton D. W.) the log shows that she was run for six hours at full speed, averaging 10.5 nautical miles per hour. Over a 7-mile stretch of this course, the speed was figured at 11.4 nautical miles per hour. At this speed her engines were making 93 revolutions per minute and indicated 1,364½ horse-power. This part of the trial is run to test a ship's propulsive equipment and determine if she makes her guaranteed speed, which in the case of the Polias is 10.5 nautical miles per hour.

After the speed run had been made, the maneuvering powers were tested. While the ship was running at full speed she was steered through a figure 8. The log shows that the complete turn on the port wheel was made in 6 minutes 25 seconds and on the starboard wheel in six minutes. The wheel was thrown from hard over to hard over in twelve seconds, which is a very good average for a ship of this size.

Next the auxiliaries were tested. The dynamo was loaded to its maximum, the ice machine tried out, feed pumps tested, and the temperature of the feed water and stack gases taken. Both anchors were let go and hove up in fifteen minutes, and the hand-steering gear used to steer the ship; finally, the engines are reversed and the ship run full speed astern for ten minutes. With the ship going full speed ahead, reversing the engines brought her to a standstill in three minutes.

These are the standard tests made on the trial trip of any ship. Both the Polias and the Atlantus handled very much as a steel ship of similar size and lines and with the same type engines would handle. The chief difference which was noticed by everyone on the trial boards was the marked lack of vibration on the concrete ship. One other point, which was found in both concrete ships, is that they backed nearly straight and can be steered slightly while backing. A steel ship with a right-handed engine always swings her stern to port when backing, and usually does not answer to her helm. Why the concrete ship should show this difference is not known, and it may not hold for the other ships, but it is a point common to both of the Emergency Fleet Corporation's concrete ships which have so far been tested.

COST AND PROGRESS RECORDS.

Weekly progress and cost reports were received from all concrete shipyards. A card of accounts was prepared upon which the cost was reported segregated in fifteen main items. The report of progress and cost includes a statement of the direct labor charge on each of these items, the man-hours of work done on each item for the week covered by the report, and the quantity of work done on each item. In addition to the fifteen items of hull construction, report is also made on the distributed prehandling expenses of chief materials entering into the construction of the hull. The setting of propellers, line shaft and rudders is also covered by this report, for although this work is really part of the outfitting, a large proportion of it may be done while the hull is still on the ways.

Each item of construction has been analyzed on the basis of total man-hours necessary for its completion and a comparative value set on each, so that the total of the items adds to 100. If the figures showing the percentage of each item erected is multiplied by this so-called "unit value" and the product of these terms are added, the sum will be the percentage complete of the entire hull.

TOTAL AND UNIT COST.

At the time of writing, total costs are available on the hulls of four 7,500 D. W. T. reinforced concrete oil tankers. The net cost for the hull alone ranges from about $600,000 to $700,000. The relative distribution of this cost for hull 1,715, 7,500-ton tanker as built by the Fred T. Ley & Company, Mobile, Ala., is shown on Fig. 43, Plate 11, no deduction being made for salvage value of materials.

As yet none of the larger ships have been completely outfitted, and no figures are now available as to the final total cost of this work. The partial figures now on hand would indicate that the outfitting will cost, including material and labor, about $600,000 for each 7,500-ton ship. The tankers will therefore cost in the neighborhood of $1,250,000, or about $167 per D. W. T. This figure should be compared with the $225 to $300 per ton cost for steel tankers built during the same period.

In Fig. 44, Plate 12, is given a time progress curve for the different items in the construction of the 7,500-ton tanker, hull 1,663, constructed by the San Francisco Shipbuilding Company at Oakland, Cal. The total time required for this ship was the shortest of any of the larger hulls so far built, but the relative proportion of the total time consumed on each item, as well as the time of starting and completing each item and the arrangement with each other, is probably typical. A time progress curve showing comparative rate of progress on each of the ships being built in the Government Agency concrete shipyards is shown in Fig. 45, Plate 13. It was originally intended that construction should proceed much more rapidly than is here indicated, and when work started on the earlier hulls the progress was greater. This extra speed necessitated a great deal of overtime work, and costs were sacrificed to speed. With the signing of the armistice the policy of the Emergency Fleet Corporation was changed, and any work being done at high cost in order to give more rapid progress was stopped, overtime work was discontinued, and lower costs made the paramount issue in construction. This naturally slowed up the rate of construction, and the curves shown are to be interpreted in this light. Another item which prevented fast construction was the inexperience of everyone concerned in the work. At Mobile and San Francisco a third ship is now being built by the same organization which have already completed two others. These are hull numbers 1,664 and 1,717. Both are now about half complete, and if the average rate of construction is maintained, they will be built in about 60 per cent of the time of the first ships at these yards.

STATUS OF CONCRETE SHIPS.

The concrete ship is a practical structure and, if properly designed and built, will resist the normal stresses to which ships are exposed at sea.

It has, however, one inherent weakness. The 4-inch shell of a concrete ship will not resist local impact of moderate intensity to the same extent as the shell of a steel ship of the same capacity. Tests were made on both concrete and steel panels designed to duplicate a section of the shell of a concrete and steel ship of similar size. These tests showed that impact loads applied between frames would only dent the shell of a steel ship and possibly loosen some rivets, while loads of similar intensity would shatter the concrete shell between frames. In cases of severe impact where the resistance of the frames is an important factor it is believed that the concrete ship will show equal, if not greater resistance than a steel ship on account of its greater mass. Up to the present writing, no concrete ships have been exposed to collision or severe impact, therefore the behavior of a concrete ship under such conditions is unknown. As a compensating feature, it has been found that concrete ship hulls are more easily and cheaply repaired than steel ship hulls.

On account of this more friable character of the shell of a concrete ship, it is necessary that it be handled more carefully than a steel ship when in harbor. It is possible some means will be found to obviate this weakness.

The future utility of the concrete ship depends upon two factors—one, its durability, and the other its economy of operation as compared with wooden and steel ships.

The final durability of the concrete ship can only be determined by years of experience, but it is quite probable that two or three years of operation of the ships now building will furnish some indication of the life which may be anticipated.

The economy of the concrete ship as compared with steel and wooden ships, assuming that it will have an equal life, depends upon the relative first cost and the difference in the cost of propulsion due to the variation in the tare weight of the hulls for equal carrying capacity. The ratio of the deadweight carrying capacity to the total displacement for the lightest concrete ship so far constructed is 0.56, which is about equal to the wooden ship and much less than the steel ship. For ships of the same dimensions the concrete ship has a greater capacity for measurement cargo than the wooden ship, and only a slightly less capacity than the steel ship. For ships of the same deadweight the concrete ship has much greater capacity for measurement cargo than either steel or wooden ships. The relative economy of the several types of ships will, of course, vary, depending upon the type of cargo handled, length of voyage, and percentage of time in port.

When the concrete ship has demonstrated its durability, many situations will be found where it will prove more economical than either wood or steel in spite of its greater hull weight, which probably can never be entirely overcome.

DISCUSSION.

The President:—Gentlemen, you have heard Paper No. 1, entitled "Methods Employed in the Construction of Concrete Ships," by Mr. R. J. Wig, Visitor. It is much to be regretted that Mr. Wig is absent, because I dare say there are a good many questions the members would like to ask. In Mr. Wig's absence the Chair will be glad to have such discussion of the paper as may be desired by the members present.

Is there any comment on the paper? Unfortunately, the paper is among those which came to hand very late. In fact, it only reached us yesterday, and I presume that very few of our members have had an opportunity to read it through. I glanced through it very hurriedly a few moments ago, and I note that Mr. Wig himself suggests the crucial test in the paragraph on page 27, in which he says: "The future utility of the concrete ship depends upon two factors—one, its durability, and the other its economy of operation as compared with wooden and steel ships." I might add a further condition, and that is its relative economy in construction as compared to the others. These are all factors whose value time alone can properly demonstrate.

The Chair is informed that Mr. Robertson might have some views to present on this subject. Will Mr. Robertson respond?

Mr. A. J. C. Robertson, *Member:*—I wish just to emphasize what the secretary has said in regard to the difficulty Mr. Wig faced in taking up the construction of concrete ships.

He started his work when every shipyard in the country was working its hardest and getting hold of every efficient man they could, and when every steel works was working its hardest to supply the needs of the shipyards, and when there was a very strong prejudice against concrete ships. He faced every difficulty with optimism; in fact, he was an extreme optimist. Many of us had criticisms of the steps he proposed, but none of us, I think, could criticise the spirit with which Mr. Wig tackled the work put before him. I think we are very much indebted to him for having tackled the job, whether he succeeded or whether he failed, and giving us a record of the result.

There is just one point I wish to correct, in Table 1. This table was prepared, of course, some little time ago, before actual measurements of the ships were possible in the matter of deadweight capacity freeboard, etc., and unfortunately the optimistic Mr. Wig was just a little too optimistic in regard to the freeboard assignment and deadweight, for the ships have not been allowed to load quite so much cargo as he claims for them.

I have no doubt that Mr. Wig will correct these figures in the final copy of his paper.

The President:—Are there any further comments? I am sure that the Society will desire that its appreciation be expressed to Mr. Wig for his admirable paper. He will also be requested to reply to the comments made and he will, doubtless, make certain additions to the paper as now presented.

The next paper on the program is entitled "Engineering Features of Shipyards Operative during the War," by Capt. R. E. Bakenhaus, C. E. C., U. S. N., Member.

Captain Bakenhaus then presented the paper.

DEVELOPMENT OF SHIPYARDS IN THE UNITED STATES DURING THE GREAT WAR.

By Captain R. E. Bakenhus, C. E. C., U. S. N., Member.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

It became a national policy to build an American merchant marine, in order that the world's losses in tonnage from the operations of enemy submarines might be replaced as soon as possible. The building resources of neutrals and countries associated with the United States in the war were insufficient to meet the demand. The United States possessed capital and resources in raw materials and a very excellent nucleus of shipbuilding plants, as well as a limited number of mechanics skilled in shipbuilding, and was in favorable condition to assume this portion of the world's burden.

The problem of producing the ships consisted of four principal elements: the financial, the developing of much additional shipbuilding labor, the supplying of shipbuilding materials, and the providing of plants for ship construction. The expansion of the Navy to meet the urgent war needs had absorbed practically all of the shipbuilding capacity of the country which was not in use for the construction of merchantmen on foreign or domestic order. The Emergency Fleet Corporation was, therefore, confronted at once with the problem of providing new plant facilities.

Before the beginning of the great war—that is, in 1912-1913—there were in the United States 49 shipbuilding yards with an estimated number of 184 ways. Of these yards, 25 were on the Atlantic coast, 8 on the Pacific coast, 16 on the Great Lakes, and none on the Gulf. Early in 1917, before the United States declared war on Germany, the number of shipyards had increased to 132, with approximately 419 ways. Of these yards, 66 were on the Atlantic coast, 32 on the Pacific coast, 27 on the Great Lakes, and 7 on the Gulf coast. By the fall of 1918—that is, before the armistice was signed and before any of the war yards were dismantled—the number had increased to a total of 243 yards, with approximately 1,202 ways, of which yards 117 were on the Atlantic coast, 64 on the Pacific coast, 29 on the Great Lakes, 32 on the Gulf coast, with one minor inland yard on the Cumberland River in Tennessee, with two ways. These numbers include yards in which the Emergency Fleet Corporation had no interests, as well as those in which the corporation's vessels were built. For convenience this information is shown in tabular form, as follows:—

TABLE I.

	1912–1913		April, 1917		November, 1918	
	No. yards	No. ways	No. yards	No. ways	No. yards	No. ways
Atlantic Coast						
1. Maine	2	6	9	19	16	53
2. New Hampshire	0	0	0	0	2	17
3. Massachusetts	3	11	4	12	6	41
4. Rhode Island	1	2	2	4	2	8
5. Connecticut	1	1	3	9	6	27
6. New York	5	22	13	53	18	85
7. New Jersey	2	12	5	29	11	95
8. Pennsylvania	3	18	4	23	7	87
9. Delaware	3	13	4	19	5	21
10. Maryland	3	11	12	40	16	82
11. Virginia	1	7	2	8	7	29
12. North Carolina	0	0	1	2	3	8
13. South Carolina	0	0	0	0	1	5
14. Georgia	0	0	2	5	7	27
15. Florida	1	2	5	14	10	39
Totals	25	105	66	237	117	624
Gulf Coast						
1. Florida	0	0	2	4	4	15
2. Alabama	0	0	2	2	6	17
3. Mississippi	0	0	1	2	4	14
4. Louisiana	0	0	0	0	7	27
5. Texas	0	0	2	3	11	53
Totals	0	0	7	11	32	126
Pacific Coast						
1. Washington	2	7	11	32	24	120
2. Oregon	0	0	9	27	17	73
3. California	6	22	12	37	23	104
Totals	8	29	32	96	64	297
Great Lakes						
1. New York	3	5	3	5	3	11
2. Ohio	4	14	5	16	5	35
3. Indiana	0	0	0	0	0	0
4. Illinois	2	6	2	4	2	10
5. Michigan	5	20	8	28	10	48
6. Wisconsin	2	5	8	21	8	40
7. Minnesota	0	0	1	1	1	9
Totals	16	50	27	75	29	153
Inland						
1. Tennessee	0	0	0	0	1	2
Grand Total	49	184	132	419	243	1202

TABLE II.

	April, 1917				November, 1918			
	Builds 3,000 D. W. T. and over		Builds ships under 3,000 D. W. T.		Builds 3,000 D. W. T. and over		Builds ships under 3,000 D. W. T.	
	No. yards	No. ways	No. yards	No. ways	No. yards	No. ways	No. yards	No. ways
Atlantic Coast								
1. Maine	3	8	6	11	8	37	8	16
2. New Hampshire	0	0	0	0	2	17	0	0
3. Massachussetts	2	9	2	3	2	22	4	19
4. Rhode Island	0	0	2	4	0	0	2	8
5. Connecticut	3	9	0	0	6	27	0	0
6. New York	5	18	8	35	8	27	10	58
7. New Jersey	2	16	3	13	7	79	4	16
8. Pennsylvania	3	21	1	2	7	87	0	0
9. Delaware	3	13	1	6	4	17	1	4
10. Maryland	2	9	10	31	6	28	10	54
11. Virginia	1	7	1	1	4	15	3	14
12. North Carolina	0	0	1	2	3	8	0	0
13. South Carolina	0	0	0	0	0	0	1	5
14. Georgia	0	0	2	5	5	21	2	6
15. Florida	3	8	2	6	8	27	2	12
Totals	27	118	39	119	70	412	47	212
Gulf Coast								
1. Florida	0	0	2	4	1	3	3	10
2. Alabama	2	2	0	0	5	13	1	4
3. Mississippi	0	0	1	2	3	12	1	2
4. Louisiana	0	0	0	0	4	21	3	6
5. Texas	0	0	2	3	9	47	2	6
Totals	2	2	5	9	22	98	10	28
Pacific Coast								
1. Washington	11	32	0	0	24	120	0	0
2. Oregon	9	27	0	0	17	73	0	0
3. California	11	33	1	4	21	98	2	6
Totals	31	92	1	4	62	291	2	6
Great Lakes								
1. New York	3	5	0	0	3	11	0	0
2. Ohio	4	14	1	2	4	33	1	2
3. Indiana	0	0	0	0	0	0	0	0
4. Illinois	2	4	0	0	2	10	0	0
5. Michigan	4	18	4	10	5	32	5	16
6. Wisconsin	3	6	5	15	4	17	4	23
7. Minnesota	1	1	0	0	1	9	0	0
Totals	17	48	10	27	19	112	10	41
Inland								
1. Tennessee	0	0	0	0	0	0	1	2
Grand Total	77	260	55	159	173	913	70	289

Table III shows the allocation of the shipbuilding ways of the country when the Emergency Fleet Corporation was formed, giving the numbers assigned to steel and wooden ships of 3,000 tons D. W. C. and over, and the number for barges and tugs and small wooden and steel vessels under 3,000 tons D. W. C. It is to be noted that the Navy occupied 40 of the ways for steel ships of over 3,000 tons capacity. The Navy also occupied, as based on the most reliable estimates obtainable, 70 per cent of the building capacity of the ways for steel ships of less than 3,000 tons capacity. Of ways not occupied by the Navy, 90 per cent were occupied by keels under foreign and private domestic contracts. This gives an idea of the very limited facilities available for the beginning of the Emergency Fleet Corporation program.

TABLE III. —*Shipyards in April, 1917.*

Yards for steel ships over 3,000 D. W. T.		Yards for wooden ships over 3,000 D. W. T.		Yards for barges and tugs and small wooden and steel vessels under 3,000 D. W. T.	
No. yards	No. ways	No. yards	No. ways	No. yards	No. ways
44	158	33	102	55	159

It is interesting to note the estimated value of the various types of yards engaged in building ships for the Emergency Fleet Corporation. The total approximate estimated value of all the yards is $369,125,000; of this sum, the amount furnished from Emergency Fleet Corporation funds is approximately $220,973,000. It is impossible to estimate at this time the returns to the Fleet Corporation from the sale and salvaging of the various properties in which it has invested capital. Table IV shows in summary form the extent of Emergency Fleet Corporation participation in yards holding their contracts.

The problem of expanding the shipbuilding facilities was solved by enlisting the initiative of existing shipbuilding companies in adding to their yards, and further by encouraging private enterprise to enter the shipbuilding business. This was, in general, done by negotiating contracts on terms sufficiently favorable to encourage enterprise. It must not be overlooked, however, that shipbuilders of the country, as well as the owners of other industries, were more than willing to contribute all their energies to assist in winning the war.

The financing of shipyard plant construction was done through various forms of clauses embodied in the contracts for ship construction. The early contracts in general provided that the Fleet Corporation furnish funds for the construction of ways and other shipyard facilities, which were generally of the most temporary character. In other cases the Fleet Corporation made advance payments with the understanding that the contractors had permission to use the advances in constructing shipyards, the amounts of such advances being deducted from payments subsequently becoming due on ships. In these cases the cost of the yards was absorbed in

TABLE IV.—*Values of Yards that Have Held E. F. C. Ship Contracts.*

Shipyard Plants

Steel yards		Wood yards		Concrete yards		Barge and tug yards		Composite yards	
No. yards	No. ways	No. yards	No. ways	No. yards	No. ways	No. yards	No. ways	No. yards	No. ways
85	453	87	361	7	12	57	165	2	8

Approximate Value of Yards Having Emergency Fleet Corporation Contracts

Steel yards	Wood yards	Concrete yards	Barge and tug yards	Composite yards
$328, 714, 000	$30, 811, 000	$5, 607, 000	$3, 293, 000	$700, 000

E. F. C. Funds Used for Plant Construction

Steel yards	Wood yards	Concrete yards	Barge and tug yards	Composite yards
$201, 435, 000	$13, 436, 000	$5, 462, 000	$631, 000	$10, 000

Per Cent of Moneys Advanced by E. F. C. for Plant Construction as Compared to Approximate Total Value of Plants

Steel yards	Wood yards	Concrete yards	Barge and tug yards	Composite yards
61	43	99	19	1

the cost of the ships In some instances loans were made and secured by mortgages on the property. In order that the resources of the country might be more fully utilized, the Fleet Corporation decided upon the establishment of the so-called agency yards. Agency contracts were made with firms which had been successful in shipbuilding or allied enterprises, to construct shipyards and subsequently the ships themselves. The yards were constructed directly from funds furnished by the Emergency Fleet Corporation, each expenditure being subject to supervision and approval in detail by representatives of the Emergency Fleet Corporation's organization.

Inasmuch as all other contracts for shipbuilding were the result of negotiations between the contractors and the Fleet Corporation, the contracts varied considerably in their terms, and it is impossible to do more than classify them in a general way. The designing, laying out and constructing of the various yards were left to the contractor who was to build the ships. He was held responsible by the Fleet Corporation for producing results. In many cases, however, contractors were required to submit their yard plans for approval by the Shipyard Plants Division of the Home Office in order that the corporation might have opportunity to examine the plans and make suggestions leading to their improvement which would increase the

Table V.—*Agency Yards.*

Location of yard	Type of yard	No. ways	By whom built
Philadelphia, Pa. (Hog Island) ..	Steel ships	50	General Contracting and Engineering Corporation
Newark, N. J..................	Steel ships	28	Builders of submarines, subchasers, and pleasure boats
Bristol, Pa.....................	Steel ships	12	Shipbuilding Corporation
Jacksonville, Fla................	Concrete ships	2	General contractor
Mobile, Ala............... ..	Concrete ships	2	General contractor
Wilmington, N. C............	Concrete ships	2	General contractor
Oakland, Cal.....	Concrete ships	2	Concrete shipbuilders
San Diego, Cal.	Concrete ships	2	General contractor
Wilmington, N. C......	Steel ships	4	General contractor

efficiency of operation. Had unlimited time been available to establish the various new yards, better results would undoubtedly have been obtained by a more extensive central supervision of plant construction, but under war conditions this was inadvisable; and, without doubt, quicker results were obtained through the policy pursued. Numerous errors in location of yards and in yard construction crept in, which no doubt interfered with certain of the project, but, as compared with the total number of yards, the errors made were comparatively few in number and were not as harmful as would have been a policy entailing any delay in the beginning of shipyard construction. The problem of getting large numbers of yards started will be described more in detail, and the elements involved in the location and construction of yards will be briefly outlined. Some of the points covered herein may seem and are elementary, but the experience during the war period has shown that even these points are likely to be overlooked or have their importance minimized.

A ship combines in itself more functions than are to be found in any one structure on land. It serves as a storehouse for all kinds of commodities and must be made adaptable to all their peculiar necessities. It serves as a carrier and must therefore be provided with means of propelling itself from place to place. It serves as a home, for a time at least, for those who operate it, and for the passengers whom it may carry, and must, therefore, be provided with all things necessary to the existence and comfort of civilized man—a traveling hotel, in fact. It is subject to all the dangers and hazards to which a land structure is subjected, besides those special ones peculiar to the sea. It must be provided with protection against all these hazards, and with the additional protection made necessary by having to fight them alone. A shipyard adapted to the peculiar needs of ship construction therefore combines in itself many more functions than are to be found in any other type of manufacturing plant. Consequently, the planning and construction of a shipyard cover more problems than are encountered in any other type of manufacturing plant.

The first question that comes up for decision when a yard is to be built is its location. The requirements for an ideally located yard are many, and it has been found in actual practice that there are few sites, if any, which ever embody all of the essentials. The primary requirement is, of course, that the yard must be on the water front. Surprising as it may seem, there were abortive attempts to build shipyards inland so that ships could not reach water except by dredging a considerable amount of land. Not only must the shipyard be built on the water front, but there must be channels which will allow the ships built to be floated to deep water. Out of the hundred odd shipyards which were built during the war it did happen that, during the early period, several were so located that their ships could not reach deep water without expensive dredging. It was incidents like these that emphasized the importance of at least some supervision of shipyard location and construction on the part of the Emergency Fleet Corporation. It is an actual fact that many sites were offered to the Emergency Fleet Corporation where conditions were ideal, excepting that the sites were too far away from the navigable channels, and it would have required excessive dredging to get the ships out and to maintain the egress channel.

It is of vital importance that the foundation conditions for shipyards be good, as otherwise excessive costs result. The number of new shipyards required was large and the time for constructing them short. It was necessary, in order to avoid delay, to make quick decision on location, and oftentimes foundation conditions were outweighed or overshadowed by what seemed more important considerations at the time. The difficulties of this character which presented themselves when construction work was well under way were overcome and the work continued to completion. Aid was rendered by the Emergency Fleet Engineers in overcoming these difficulties. As an illustration a certain yard had proceeded with its construction when reports were made that the foundation conditions were bad and that the building of the yard should stop. However, an engineer from the Shipyard Plants Division visited the site, a solution was found, and the work continued.

Very frequently, after a site was found that met all of the requirements outlined, it proved to be too far away from the labor market. The valuable water front property near large cities was in general occupied by other enterprises, and thus new shipyards were frequently forced to locate away from the heart of industrial centers. Manifestly this introduced problems in housing and transportation of shipyard employes. The Housing and Transportation Division was at one time a section in the Shipyard Plants Division, but so great did the problem become, on account of impossibility of locating sites near industrial centers, that it was decided to establish a separate division for this phase of the work. The importance of the work may be indicated by stating that this new division constructed many transportation lines and a great number of housing communities, at the cost of many millions of dollars.

Climatic conditions are particularly important in shipbuilding, as a large part of the work is done out of doors. It is sufficient to say that so great was the demand for ships that the climate, though deterrent, was not found to be a determinative feature. In the case of the five concrete yards, however, climate was the deciding factor in locating the yards where the winters were decidedly mild.

In selecting the character and type of construction to be used in the new shipyards, an effort was made to determine the possibility of their use after the war. Some of the yards were treated as existing only for war emergency, and the character of construction selected provided only for the life of the ship contracts awarded. Others were constructed more permanently with the idea that they would continue in operation indefinitely. Still others were constructed with the idea that they would continue to operate on a reduced scale, after the completion of their war contracts, and that certain alterations would be made to enable them to compete under commercial conditions.

The shipyards in which steel, wooden or composite ships were to be constructed were fairly well determined as to type and equipment by previous experience. Effort then was made to improve the design of these yards by studying their construction methods and introducing new types of machinery and equipment of such character as would increase their output per way over pre-war standards.

The concrete shipyards presented special problems which were so unusual in shipbuilding that an entirely new type of building talent was utilized. The contracts for building the concrete ships were, in fact, given to firms which had had no experience in shipbuilding, but which were experienced in reinforced concrete work. The only activity adaptable from the hull division of the ordinary shipyard was the mold-loft. The forms for molding the concrete involved carpenter work of a very high grade and necessitated employment of mechanics accustomed to form work, that is, who could build a structure "inside out." The remainder of the work involved the handling of concrete materials and the placing of reinforcing steel. It is clear that none of the ordinary shipyard machinery for either wood ships or steel ships could be utilized.

The Emergency Fleet Corporation established five concrete shipbuilding yards at various points. It was unfortunate from the technical standpoint that the number of ships originally contemplated was greatly curtailed because of the ending of the war, and that the resulting data as to costs of building concrete ships will not represent the lowest cost obtainable. Within limits, the concrete floating craft, whether in the form of ships, lighters, barges, pontoons or floats, undoubtedly has a future, and the importance of the experience of the Fleet Corporation in developing this type cannot be too greatly emphasized.

The ways upon which a ship is built perform a double function. The building ways are a foundation for the ship while it is a land-borne structure, and as soon as it is sufficiently far advanced the launching ways provide the means for transferring the hull to the water. Necessarily a part of the launching ways must be built below the surface of the water. The ways structure depends first on the class of ships to be built and, second, on the foundation conditions. In the great majority of instances the ways are built on wooden or concrete piles. In a number of cases the ways, or parts of them, are built on concrete or wooden foundations carried below the frost line to hard ground.

The old line shipyards had developed ways which had proven satisfactory in operation, and these yards needed no supervision from the Fleet Corporation in this regard. A large number of new yards, however, had had no experience in ship construction, and a certain amount of supervision and advice in these cases was not only advisable but practically obligatory if the interests of the Fleet Corporation were to be protected and ships produced at the rate demanded by war conditions. For this reason the Shipyard Plants Division prepared standard drawings of shipbuilding ways for various types of vessels based upon calculations made in the division from the ship plans furnished by the Ship Construction Division.

Ways are of two distinct types, end launching and side launching. The end-launching ways are the customary type on the seaboard, whereas side launching is used almost exclusively on the Great Lakes. One of the results of shipbuilding work during the war was to increase the number of side-launching ways on the seaboard. The five concrete shipyards were constructed with side-launching ways.

Dredging at shipyards is required for providing launching basins, installation berths, channels to deep water and for filling low lands of the yard. The importance of this work may be indicated by the statement that dredging operations conducted under the cognizance of the Emergency Fleet Corporation totaled 34,300,000 cubic yards at a cost of $9,400,000. The Dredging Section of the Shipyard Plants Division exercised supervisory control over dredging operations for 141 shipyards, marine railways, dry docks and installation plants, in 127 of which the Emergency Fleet Corporation was directly interested financially by loans, investments or otherwise. In this work, as in nearly all of the other work of the corporation, the initiative was left with the shipbuilder, but central control was necessary to a much greater extent, because of the inter-relations of this work with that of the War Department, the Navy Department, the United States Railway Administration and others, and because the shortage of dredging equipment necessitated careful pooling of all available dredging outfits. At times the demand for dredging equipment was so great that most careful consideration was necessary in distributing equipment, in order that the interests of the Government, as a whole, might be best served.

In but three cases have the vessels been delayed on account of lack of dredging. In two of these cases the financial condition of the contractors was such that no dredging company could be found to do the work until payment was guaranteed by the Emergency Fleet Corporation, and in the remaining case the dredging, which was under cognizance of the War Department, could not be done on account of unfavorable weather conditions.

The function of the Dredging Section was to determine dredging requirements and to supervise the work of dredging, so that the launching and delivery of vessels might not be delayed. The providing of dredges to accomplish this purpose, the keeping of records of all dredges in the United States, including their type, capacity and the work on which they were engaged, the settling of disputes arising over dredging operations, the securing of surveys of work accomplished, the personal inspection of dredging operations and the keeping of records of the requirements, progress and completion of all dredging, were details incidental to this work.

Railroad connections and yard railroad systems are important features of shipyards. Some of the shipyards constructed for war purposes were of such size that the problem of handling incoming and outgoing cars was of vast importance. It involved the design of complete railroad yards within the shipyards themselves, which included all the necessary trackage for handling quickly and economically the number of cars received daily, necessitating receiving tracks, classification tracks, storage tracks, interchange tracks and the adequate connections and auxiliaries required for such development. In addition, there were provided the tracks leading to the storage yards, the shops, the ways and the outfitting piers. To handle the materials within the shipyard limits there were provided locomotives, locomotive cranes and cars together with repair facilities and housing facilities for the transportation equipment. All this detail was rendered more important by the fact that the railroads of the country were strained to the utmost to meet the conditions imposed upon them by the war, and the rapid handling of cars was no small factor in aiding them to meet their problems successfully.

The importance of sanitation systems was early recognized in the Fleet Corporation and a Department of Health and Sanitation was established in the Shipyard Plants Division, having at its head an officer of the Medical Corps of the Army, assisted by a staff of sanitary engineers and physicians. It was felt that the application of modern sanitary methods and standards to the shipyard could not be left to private initiative. The yards did not ordinarily possess the talent required for initiating this kind of work. It is true that, in some of the states, the sanitary regulations were such that local communities had been encouraged to establish satisfactory sanitary standards. In general, however, the health of the workman and his safety, from a sanitary standpoint, required active steps on the part of the central organization of the Fleet Corporation. After the Department of Health and Sanitation had been well established and arrangements made to provide physical equipment, it was determined that the department should be transferred from the Shipyard Plants Division to the Industrial Relations Division because of the great importance of the control of the personnel in the shipyards.

The safety of the men was dependent not only upon proper measures as to health and sanitation, but also upon the provision of modern safety methods of construction and operation. A safety section was established in order to insure proper forms of construction and the guarding of machines and dangerous places, so as to eliminate avoidable accidents. This section undertook to further reduce accidents by instilling into the workmen habits of care and caution in conducting their work. The Shipyard Plants Division cooperated in the establishment of this work, which was placed under the Industrial Relations Division because of the greater facility in reaching the men. The Shipyard Plants Division further cooperated with the safety section in the physical side of this work, lending aid by constructing such mechanical protection as was required and by incorporating into plant construction the best ideas on this subject.

Plant protection was instituted as a function of the Shipyard Plants Division and was accomplished through guarding the yards with civil watchmen and military forces, as well as with adequate fire protection. The plant protection work was later made to include information service as to irregularities, and was on that account transferred to the office of the Director General.

It will be recalled that, in the fall of 1917 and during the early part of 1918, the country was very apprehensive as to the danger from fires in the shipbuilding plants and in other vital war industries. This led the Emergency Fleet Corporation to make strenuous efforts to keep down fire losses. To further this work, immediate steps were taken to establish fire protection systems and careful plant guarding. The remarkably low losses from fire, while in part possibly due to good fortune, may, nevertheless, be attributed to the effective work done by the corporation in guarding against fire. The following table from the fire protection report shows the salient facts:—

(*a*) Total number Fleet Corporation Yards covered, 186.

(*b*) Total number of installation yards covered, 19.

(*c*) Total value of shipbuilding property and ships protected (estimated), $1,500,000,000.

(*d*) Total fire losses (above $1,000 each), $305,150.

(*e*) Percentage of loss 2/100 of 1 per cent or $1.00 lost for each $5,000 invested.

(*f*) Total value of insured property in entire country, $100,000,000,000.

(*g*) Total fire loss in entire country, $290,000,000.

(*h*) Percentage of loss 3/10 of 1 per cent or $1.00 loss for each $333 invested.

While the various estimated amounts may not be absolutely correct, it is nevertheless of interest to know that the closest obtainable estimates indicate that the loss in property under Fleet Corporation supervision was only 1/15 of that occurring in property of the United States generally. If the proportionate loss under the Fleet Corporation supervision had been the same as existing in the country generally, the loss would have been $4,800,000 instead of $320,000.

Fire protection measures were applied through a section established by the Shipyard Plants Division of the Fleet Corporation. The section was mainly composed of engineers supplied through the courtesy of the National Board of Fire Underwriters. In addition to the Home Office force, field work was done in part by traveling engineers from the Home Office, but principally by the local agencies of the National Board of Fire Underwriters. All shipyard and installation plants and other establishments doing work for the Fleet Corporation were given a thorough inspection by trained experts, who reported on the adequacy of their fire protection systems. These reports were sent to the Home Office and steps were taken to install proper fire protection systems where none existed, and to bring others to a satisfactory standard. Layouts of fire protection piping were furnished to the shipyards and fire engines and other equipment purchased and properly distributed. The financial question immediately confronted the Fleet Corporation. Many of the yards claimed that they did not need fire protection to the extent demanded by the corporation. Others claimed that they were without funds to establish the systems. In one case, a yard maintained that it was not in need of fire protection, having existed for half a century without such protection. That particular yard was burned to the ground a few months after its assertion had been made, and this experience had a decided effect on other yards. It became firmly established that the only way in which fire-protection systems could be installed under the existing conditions was by pressure and financial aid from the Fleet Corporation.

If we bear in mind that the country wanted ships, it will be clear that the Emergency Fleet Corporation was justified in financing fire-protection systems. It was felt that not even the loss of a single vessel should be incurred if it could be prevented by the expenditure of the comparatively small amounts invested in fire protection. It is true that the vessels were covered by insurance, but money paid as insurance in case of fire could not carry goods to Europe. It was pointed out particularly that the Navy was spending $500,000,000 for the construction of destroyers whose function was to prevent the loss of merchantmen through submarines. Fire protection systems performed a similar function, and the $2,000,000 requested for the installation of fire protection did not seem excessive.

For the twelve months ending February 28, 1919, there were 519 fires in shipyards; 372 of these fires caused no loss, while 157 caused a total damage of $320,000. During this period 1,805 keels were laid, 1,040 hulls launched, and 625 ships delivered. Only four hulls were damaged by fire, causing a total loss of not more than $50,000. Only two fires caused a loss exceeding $100,000. No fire protection had been installed in either of these yards, and our reports show that had water mains and other fire equipment been in service, neither of these yards would have been badly damaged.

With the designs of the shipyards prepared under war conditions by many different interests, some of whom had had but little experience in shipyard work, it can readily be understood that the results were not always the same. A comparative study has been made of the facilities which were provided: First, that of all yards under the cognizance of the Emergency Fleet Corporation; second, that of the yards showing the best production. The results of these examinations of the Emergency Fleet Corporation records are given in the following discussions:—

The measure of the efficiency of the yards included in this study is taken as their yearly output in ships. For this output study the steel shipyards were divided into certain classifications according to their ability to build ships within limits of length and breadth, and for the purposes of the study were grouped into two divisions. The first division consists of Classes B and C, and includes ships from a minimum length of 175 feet and a minimum breadth of 36 feet to a maximum length of 345 feet and a maximum breadth of 52 feet. The second division consists of Classes D, E, and F, and includes all ships larger than 345 feet in length and 52 feet in breadth. Plants included in the second division are building vessels of 7,500 deadweight tons or greater. Plants have been given arbitrary numbers for convenience of reference in compiling and plotting the data.

In order to show more clearly the relative value of the work done in the plants building ships of different deadweight tonnage, the yards are classified in order of equivalent deadweight tonnage. It will be readily understood that the completion of three 3,500 deadweight-ton ships is a greater undertaking than the completion of one 10,500 deadweight-ton ship, although the actual tonnage is the same. Therefore a table of coefficients previously adopted and in use for just such purposes was used and applied to the actual deadweight-ton figures in making a comparison of the output of the yards building different sizes or types of ships. In this table a 7,500 deadweight-ton cargo ship has been used as a standard, with a coefficient of one. The deadweight ton and the equivalent deadweight tons (E. D. W. T.) of the 7,500-ton ship are therefore the same. The coefficient for a 3,500 deadweight-ton cargo ship is 1.5, and therefore the E. D. W. T. for three 3,500 deadweight-ton ships on this basis would be three times 3,500 times 1.5 or 15,750 equivalent deadweight tons. The coefficient for a 10,500-ton cargo ship is 0.81 and the E. D. W. T. for this ship would therefore be 8,505.

By making this transposition of the yards considered, the E. D. W. T. becomes the standard of measurement of output for the purpose of relative comparison of the forces necessary to produce that output. All plants considered have been compared on the basis of average and maximum E. D. W. T., average and maximum steel tonnage, and average and maximum deadweight tons.

Chart A, Plate 14, shows the average and maximum output per annum of Classes B and C plants in tons of steel, in deadweight tons and equivalent deadweight tons, the plants being arranged on the left of the chart in the order of maximum equivalent deadweight tons, the largest first. The number of ways used in obtaining the output are also shown on the left of the charts. On the right the plants are arranged in order of maximum equivalent deadweight tons per way, the largest first. In the center of the chart are arranged the six leading yards, plotted on the basis of the average E. D. W. T. per way. For the purpose of comparison an average line representing the total output of yards with from three to ten ways has been plotted, and the actual total output of the six yards compared has been plotted against this line.

Chart B, Plate 15, shows the same data except that the plants have been arranged in the order of the maximum tons of steel produced. In the center of the chart are shown the six plants leading in the production on a steel tonnage basis.

Chart C, Plate 16, shows the second division of plants arranged in the order of the maximum equivalent deadweight tons with maximum production per way on the right. In the center the twelve plants leading in maximum equivalent deadweight tons per way are compared separately. Such a marked difference appears between the performance of the four leading plants and the other eight that it is impossible to make a comparison of the actual output per way as was shown on Charts A and B. Instead three averages have been plotted, one showing the leading four plants, another the remaining eight, and the third the average of the twelve.

Chart D, Plate 17, shows the second division plants arranged in order of maximum steel production. The per way averages of the twelve yards leading in steel production are plotted on the same basis as those on Chart C, Plate 16.

In studying the plants it was necessary to compare them on the basis of their various functions, and the divisions used in this study are listed below. They have been selected as being the most natural divisions into which the work of building ships can be divided and are common to all shipbuilding plants. These in turn may be further subdivided, but, from lack of data systematically gathered, it is not possible to go into greater detail at this time.

Hull factors.	*Outfitting factors.*
Ways.	Berthing space.
Plant area.	Machine shop.
Storage area.	Smith shop.
Fabricating area.	Woodworking shops.
Assembling area.	Stores.
Mold-loft area.	Combination shops.
Tracks.	Miscellaneous buildings.
Punches.	Boiler shop.
	Foundries, iron and brass.

The following is an explanation of the Hull Factors used for making this study:

Number of Ways.—This factor represents the number of places in the plant where hulls have been or may be constructed of the size and tonnage reported in the output for that plant.

Plant Area.—This factor is in acres and represents the actual area used in conjunction with the number of ways listed, deducting all open water and including all wharves and piers.

Storage Yard.—This factor is in square feet and is the actual area used for the storage of ship stock which consists largely of plates and shapes and is usually uncovered, including the tracks which serve the same. The principal function of this area is to act as a reservoir to maintain a constant supply of materials to the fabricating area.

Fabricating Area.—This factor is in square feet and includes the actual floor area of shops used in the working of plates and shapes. It includes the areas used for punching, shearing, bending and anglesmithing, and the areas of bending slabs, and furnaces. In yards where some of this work is done out of doors, the necessary space has been included. There has been excluded, as far as possible, all assembling space.

Assembling Area.—This factor is in square feet and is the area, usually adjacent to the fabricating shop and ways, used for bolting up and riveting of hull parts and for storage of finished steel pieces and fabricated material (sometimes called the fabricated steel stores) held ready for erection. Where assembling is a part of the fabricating shops, an effort has been made to separate the two classes of work. This space has the important function of equalizing the supply and demand between the fabricating shops and the ways.

Mold Loft.—This factor is in square feet and represents the floor area used for developing the ship's drawings and the manufacture of templets or full-sized patterns by which the various plates and shapes are fabricated.

Railroad Tracks.—This factor represents the actual lineal feet of the standard gauge track used in the operation of the plant, exclusive of the connection to the main line usually outside the main plant.

Punches.—This factor represents, so far as possible to obtain from the records, the number of punches used in the fabrication shops. These, with the shears, are the principal tools, but, owing to the common practice of having interchangeable punches and shears, the record of the number of shears, to obtain which an attempt was made, was of little value, and some of the punches shown may be at times used as shears.

The following is an explanation of Outfitting Factors used in making this study:

Machine Shop.—This factor is in square feet and is the actual floor area, including balconies and second or other floors used for machine shop purposes, such as making of machinery, machining of hull parts, tool repair, making of templets, gauges, jigs and dies. The records show that all plants have such facilities in greater or less degree. A small shop is necessary even if the heavy machinery is purchased outside. Some plants manufacture all of their equipment, and some have surplus capacity and manufacture for others.

Smith Shop.—This factor is in square feet and is intended to include the blacksmithing work of the plant, all drop forgings, all hammered work, forgings for ship and engine parts, repairs, manufacture of templates and machine tools. The size of this shop depends largely upon how much of its machinery the plant manufactures. Where very little is manufactured, this shop is often combined with the anglesmithing. There is some conflict in the records on this account. There are also plants that have surplus capacity and manufacture for others.

Woodworking Shop.—This factor is in square feet and includes all saw-mills, joiner shops, ship carpenter shops. The ship carpenters or shipwrights place blocking, scaffolding, decks, hardware, etc., and manufacture masts, booms, and spars. It also includes pattern shops used for the manufacture of all wood patterns required for the casting of iron, steel or brass. Many plants without foundries manufacture their own patterns. This item could be subdivided into several branches, but the necessary information is not available.

Stores.—This factor is shown in square feet and represents the actual floor area of buildings used for storage purposes.

Combination Shops.—This factor is in square feet and includes all marine machinists and marine electrical shops, all pipe, copper and sheet metal shops. This item is capable of being subdivided into several branches, but the information is not available.

Miscellaneous Buildings.—This factor is in square feet and is the floor area of the many buildings that do not come in the above divisions, excluding the main office building and restaurant, barracks and hospitals where they exist.

Boiler Shop.—This factor is in square feet and is the floor area of buildings used for the manufacture of boilers. Some yards have no such shops, while some have shops with surplus capacity. The latter are noted so far as possible.

Foundries.—This factor is in square feet of floor space. Comparatively few plants have complete foundries for all classes of work. Some have brass only, and a few have foundries with surplus capacity. These have been noted as far as possible.

The plant factors have been plotted for convenience of comparison. Those for Classes B and C are on Chart E, Plate 18, and those for Classes D, E and F on Charts F and G, Plates 19 and 20. These charts make it possible to quickly compare the make-up of the several plants and to draw general conclusions. The plants have been assembled in order of the number of ways, with the largest first. In general, the several factors appear to have but little relation to the number of ways in the plant, there being a great variation in size of the factors for plants with equal number of ways. A few of the factors have fairly well-defined average increases with the increased number of ways. Others do not. Several appear to be of the same size for four, five and six ways. The factors for machine shop, smith shop, boiler shop and foundries are shown as they have been obtained, but owing to the difference in the extent to which they are used in many plants, and their absence in others, it is difficult to draw definite conclusions. Where plants are known to be deficient in capacity or have a surplus capacity, this fact has been so noted on the charts.

A careful examination of the data which has so far been prepared indicates very little uniformity in the extent of equipment provided for the various yards. Even in the yards of greatest efficiency, as indicated by their output, there is very little uniformity. This leads to the conclusion that the efficiency of a shipyard is less dependent upon plant layout than upon other factors, the principal one of which is, of course, the personnel. There is no doubt, however, that while plant layout is not the most important factor, it is nevertheless one of great importance in the establishment of a building yard. The study of a relation of plant layout to efficiency in plant output has not been contemplated in time to be outlined in this paper. It is therefore not safe to draw any conclusions at this time as to the exact function of plant layout in determining efficiency.

The Shipyard Plants Division of the Emergency Fleet Corporation was organized under Admiral H. H. Rousseau, who remained as manager of the division until May 1, 1919. It is desired to acknowledge the valuable aid rendered by Mr. J. E. Tonnelier, head of the Records and Progress Section of the Shipyard Plants Division, and Mr. Sherman A. Jubb, supervising engineer, in the preparation of this

article. It is only fair to state that whatever good results have been accomplished through the Shipyard Plants Division have been due to the unusually loyal and efficient service rendered by the engineers and all other employes engaged on the work.

DISCUSSION.

THE PRESIDENT:—Gentlemen, you have all heard Captain Bakenhus' paper. I am sure that we appreciate his effective presentation of this most admirable paper. It is necessarily a record of facts, but they are very illuminating facts, and in future time we will all refer to them with very great interest. Is there any comment?

MR. STEVENSON TAYLOR, *Past President*:—I have no special comment to make upon the paper other than that it seems to be a most admirable one to be presented at this time, and as you have just said, Mr. President, it is mostly a record of facts, but facts that have been put together in such a clear and concise manner that we are all ready to thank the author for the excellent presentation. It will also be interesting to have this same question followed up in another year, because probably during the coming year there will be material changes in the number of yards and number of ways and the questions of salvage which will certainly be most interesting in that time.

When it becomes in order, I should like to make a motion that the thanks of the Society be given to Captain Bakenhus for his most excellent paper.

THE PRESIDENT:—The motion is in order now.

MR. TAYLOR:—Then I make that motion.

THE PRESIDENT:—Gentlemen, you have heard the motion that Captain Bakenhus be given a vote of thanks for his excellent presentation of an important matter in connection with the establishment of shipyards.

The motion was put to vote and unanimously carried.

THE PRESIDENT:—The next paper on our program is entitled "Steel Ship Construction from a Management Viewpoint," by Mr. Creighton Churchill, Member. The paper will be presented by the author.

Mr. Churchill, previous to reading the paper, said:—"This paper is the result of a rather unusual opportunity which presented itself owing to war conditions through the Emergency Fleet Corporation; that is, we had the opportunity of making a very careful analysis of the shipbuilding industry of the country, and from that analysis we developed certain very interesting points, and among others is the subject of this paper. It is not very long, so I think the best thing to do is to read it in full."

Mr. Churchill then read the paper, and during such reading said:—"I may say that these costs as developed by the curves are more for comparative purposes than for actual costs. For instance, the matter of overhead used was purely an estimate. As you probably all know, the overhead in one shipyard and that in another are two very different things. As it worked out, what we did use was approximately correct."

The paper in full follows.

STEEL SHIP CONSTRUCTION FROM A MANAGEMENT VIEWPOINT.

By Creighton Churchill, Esq., Member.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

A matter of intense interest to the shipbuilding industry of the country is its ability to compete with foreign yards during and after the period necessary to provide tonnage to make up for war losses and the normal increase during the period of the war.

Published records show 21,800,000 deadweight tons of shipping lost since the beginning of the war, during which time 15,700,000 tons were constructed, making a net loss of 6,100,000 deadweight tons. The rate of construction for five years prior to the war was 2,300,000 tons per year, which, it is anticipated, will become about 5,000,000 tons per year until 1924 or 1925, when the rate of construction will become a matter of replacement and natural increase due to trade expansion, probably somewhere in the neighborhood of 3,000,000 tons a year. Naturally a very large percentage of this will be steel construction.

As of April, 1919, there were sixty-seven yards in this country with 425 ways capable of building steel ships of over 3,000 deadweight tons. There were under construction at these yards 637 steel vessels aggregating 4,225,000 deadweight tons, and 591 vessels, aggregating 4,200,000 deadweight tons, yet to be constructed. In this country alone, therefore, provision has been made to add some 8,425,000 tons to the world's shipping with a total yearly output of approximately 3,000,000 to 4,000,000 tons, depending on the type of ship. It is apparent, therefore, that it will take about two years to complete the present program, provided it was spread over the entire industry. This, however, is not the case, and it is estimated that at the close of the present year from 30 per cent to 50 per cent of the country's capacity will be available for contracts for private account. In other words, a capacity of from 1,000,000 to 2,000,000 tons a year will be released, and it is tonnage to this amount that must be built to meet immediate demands and at the present prices of material and labor. If this business can be secured, it is the writer's firm belief that the shipbuilders of this country will find the means to develop the industry along modern lines such that its future will be assured.

During the war, time was the very essence of all contracts, with expense a less than secondary consideration. With conditions reversed and a reasonable time to adjust themselves on a strictly economical basis, shipbuilders will continue in business only so long as they pay strict attention to factors under their control. It was hardly to be expected that any great degree of man efficiency would be shown as a result of construction during war time. In some few cases—those of old-established yards, where the percentage of new men was comparatively low—the records of performance were little short of wonderful. In a very large majority of cases, however, both the men and the management were decidedly lacking in skill and experience, which is easily accounted for if one stops to consider that in a remarkably short space of time shipyard labor increased from 50,000 to over 300,000 men. To supply competent and skilled supervision for such an increase was next to impossible, yet, in a way, it was done and ships were launched and sailed the seas to the everlasting credit of the country. "Ships and more ships" was the slogan then, and the ships came. "Low costs and a living wage" might well be the slogan of the future, for it can very easily be demonstrated that with a reasonable rate of production on the part of labor there is no valid reason why this country should not maintain its high standard of living, which means high wages, yet secure all the business necessary to keep the industry in a healthy and flourishing condition.

A cursory analysis of the situation indicates that the factors under direct control of the shipbuilders are organization, management, methods, and, to a greater or less extent, men. The prevailing market controls both material and wages, the latter through labor being considered a commodity. The spirit of the times would indicate that labor as a commodity is not in accord with the future relations between capital and labor, and it is, therefore, extremely doubtful if the market will exercise its former control on wages. For the purposes of this paper this has an important bearing in that the element of wages will be treated as one of the factors not under control by the shipbuilder. The direct effect of such treatment will be to concentrate attention on factors that, beyond any question, are under control.

It is a foregone conclusion, other things being equal, that the success of an industry is very largely a matter of organization and management. In modern thought the former is considered a science and the latter an art. Science above all things demands exactness, and the reasoning back of referring to organization as a science is that it is, or should be, based on certain facts determined entirely by the nature of an industry and the various operating functions by which it is controlled. That management may justly be termed an art must be admitted when one considers the varying degrees of success attained by different men in the same position and the same tools (the organization) with which to work. Methods are simply a means by which the artist (the management) manipulates the tools (the organization) at his disposal. In many cases, perhaps a majority, methods are more or less a growth fostered by the necessity of the moment, a sort of patch work with additions here and there to fill in a vacancy. For example, it may happen that a method of storekeeping is installed that in itself is perfectly satisfactory as far as stores are concerned, but involves other departments to their detriment. Methods should be considered from the point of view of the organization as a whole. If looked upon as the means of coördinating the activities of the various units of the organization, the problem will be much simplified. As a general rule, methods should be the result of a careful analysis, otherwise a useless amount of paper work may creep in. They are only useful in so far as they accomplish a well-defined purpose and in the simplest possible way. The writer has in mind, as an example of a method installed without having been given proper consideration, a case of keeping time to tenths of an hour without any provision having been made to provide the next job for the man. From the cost-accounting point of view this was fine, but it did not mean more accurate costs for the very simple reason that the actual costs depended entirely on how soon the foreman could find the next job for a man to work on. In other words, in this particular case, keeping time to tenths of an hour was a perfectly useless refinement, did not tell the true story and, worst of all, was in no sense a fair measure of man efficiency. Incidentally, the above method of keeping time was installed at the suggestion of the accounting department and is a good illustration of the tail wagging the dog. Had the matter been referred to a properly qualified management expert from the outside, no such half-way measures could have been the result of even a superficial analysis of contributing factors. As a general rule it has been found to be a safe policy to secure the services of such experts, when circumstances warrant, on the theory that, having a specific problem to solve and not being directly concerned with the daily activities of a going business, they have everything in their favor.

Assuming that the organization, management and methods fulfil requirements, the only remaining factor under control is the men. On this factor hinges final costs after defects in the other factors under control have been remedied. It may be of interest to know that the results of analytical studies of operations in over 50 per cent of the steel yards engaged on construction for the Emergency Fleet Corporation during the war indicate that less than 20 per cent of the faults developed were chargeable to the men. This is simply another way of saying that other factors under control were responsible for some 80 per cent. Accepting this statement as correct, it will be shown that the combined result of reducing the faults of both men and management is a very material reduction in construction costs. In fact, the practicability of so doing is what confirms the writer in his belief that the shipbuilding industry will have itself to blame if it cannot stand on its own feet in the face of foreign competition.

In investigating the costs of ships of different types and cargo carrying capacity an intimate relation developed between fabricating capacity, number of ways, construction program and man efficiency, the last expressed in deadweight tons of output per man per year based on the entire force engaged on ship construction, exclusive of office force and men engaged on plant construction. A 7,500-ton deadweight cargo vessel was used in the calculations, with material costs as follows:—

Hull, $212,952 (plates at 3¼c.; shapes at 3c.; bars at 2.9c.).

Machinery, $266,515 (2,899 horse-power reciprocating engines; three 13 feet x 12 feet Scotch boilers).

Equipment, $104,371 (military requirements, $30,000).

A certain fixed overhead per ship per day was used and due allowance made for the extra cost of military requirements to bring the final costs down to a strictly commercial basis for a ship of this type and specifications.

The labor cost was calculated on the basis of man efficiency, using $5.40 as the average daily wage of all men engaged on ship construction. To arrive at the number of men per way the following formula was used:—

$$\text{Men per way} = \frac{300 \times a}{b \times c}$$

in which

a = fabricating capacity per 8-hour day.
b = tons steel erected per man per year.
c = number of ways.
300 = number of working days per year.

Launching was assumed when 90 per cent of the steel was erected and rivetted. A balance between fabricating, erecting and rivetting was taken for granted.

Plate 21 shows deadweight-ton cost curves for a five-way yard, plotted for different building programs and man efficiencies with fabricating capacity to meet program requirements. The Minimum Cost Line shows the most economical building program for certain given conditions.

Plate 22 is a companion curve to those in Plate 21, being plotted from the same data, and shows the fabricating capacity corresponding to the most economical building program.

Plate 23 shows deadweight-ton cost curves plotted for different numbers of ways and man efficiencies with a fixed fabricating capacity of 150 tons per 8-hour day. The Standard Ways Line shows the most economical number of ways for different man efficiencies.

Plate 24 shows cost curves based on the same data as those in Plate 23, but referred to deadweight-ton cost and man efficiency as coordinates.

A careful study of Plates 21 and 23 develops some very interesting results which may be expressed as laws and enumerated as follows:—

1. For a given man efficiency there are a certain definite number of ways and a certain building program that will result in minimum costs.

2. As man efficiency increases costs decrease according to a diminishing ratio and not in direct proportion to increase of efficiency.

3. The lower the man efficiency the greater the number of ways and the longer the building program to secure most economical construction.

The intimate connection between man efficiency and building program shows, beyond any question, the advisability of working to standards and setting such standards within reach of the working force. For example, if a contract were secured for delivery of a 7,500 deadweight-ton cargo vessel in 155 days from keel laying (125 days keel to launching and 30 days launching to delivery), there is a possible variation in cost of about $117 and $158 per deadweight ton, depending on whether the yard can show a man efficiency of 65 deadweight tons or 25 deadweight tons per man per year with a fabricating capacity of 90 tons per 8-hour day. Obviously this difference is entirely a matter of labor cost and shows the necessity of paying all possible attention to man efficiency and making every effort to bring it up to a standard. It is suggested that standards be set in accordance with the following table which represents actual performance on cargo ships in going yards for an equivalent period of 120 months:—

D. W. T. tonnage	*Standard per man per year*	*Values of "b"*
3,500	30 D. W. T.	10
5,000	36 D. W. T.	12
7,500	45 D. W. T.	16
10,000	53.5 D. W. T.	20
12,000	61 D. W. T.	23

The values of *"b"* corresponding to the different man standards for ships of different tonnage is included in the above table because it is the basis for figuring the number of men per way to attain the given standard. (The values of *"b"* listed were derived from actual performance in twenty-three yards. Of these yards 37 per cent manufactured engines and boilers, 32 per cent either engines or boilers, and the balance, 32 per cent, neither engines nor boilers. This is mentioned that proper allowance may be made in the use of the formula for individual yards.)

Plates 25 and 26 show the man standards and values of *"b"* in the form of curves for cargo ships from 3,000 deadweight tons to 13,000 deadweight tons based on 45 deadweight tons per man per year, for a 7,500 deadweight-ton cargo vessel.

The main value of standards of any kind is the use to which they are put in practice. It is a very grave mistake to utilize them only for executives that they may be kept conversant with performance. As a matter of fact much better results would be obtained if they were used for the information of the men doing the work. The common-sense way of using them is to present performance referred to standards to both the men and executives. In practice, performance with reference to standards is best presented to executives on a cost basis and to the men on a productive basis.

Ship construction depends primarily on a balance being maintained between fabricating, erecting and rivetting, with outfitting and machinery installation depending on the progress of these three elements. That the standards set may be used to best advantage, Plates 27, 28 and 29 are suggested as a form for presenting performance to the men. In the writer's opinion little or nothing is accomplished by presenting mere figures, from day to day, of the previous day's performance. Graphical records are much simpler to grasp at a glance and tell the story as a picture. Owing to the natural varying rate of ship construction, from day to day, the system of averages is used.

Plates 27, 28 and 29 are based on building 7,500 deadweight-ton cargo vessels in a five-way yard with 150 tons fabricating capacity on a predetermined 75-30 building program. (Reference to Plates 21 and 23 will show that this will be the most economical program only if the man efficiency is 65 deadweight tons per man per year or over.) Average daily performance by weeks is shown graphically, while figures are used to show this average to date. As drawn up, these graphs are intended for use by the general superintendent and the general foremen responsible for the different activities. For the information of the men the same general idea can be followed out with such changes as may seem desirable. It is suggested that, for this purpose, daily performance be used in connection with blackboards at each way and in the fabricating shops.

In considering these records it will be noticed that the steel fabricated at the end of twelve weeks is 12 tons short of the required average output, that steel erected is 39 tons short and ship rivets 3,000 short. Investigation or study of the data recorded shows at once that the steel erection is not keeping pace with the fabricating, while the rivetting is being held back on account of erecting. This information in the hands of the general superintendent tells him, beyond any question, where to concentrate his attention to correct troubles or remedy faults not under control perhaps by the erecting foremen. Naturally this would be done before twelve weeks had elapsed; in fact, his attention would have been called to existing conditions after the second or third week, and the results at the end of the twelfth week would show the extent to which he had been able to correct the trouble. In actual practice, of course, it is highly probable that the erecting and rivetting records were below the standard set for well-known reasons, but the fact remains that the yard has fallen down on the program because the erectors have been unable to keep up with the fabrication, also that the daily average output of the fabricating shops was not up to standard.

As a final example of the necessity of giving consideration to all the elements involved in shipbuilding, the curves shown in Plate 30 were developed as indicative of what happens in actual practice. A five-way yard is taken with a man efficiency of 35 equivalent deadweight tons per man per year. The problem is treated from two points of view, namely:—

1. Constructing cargo vessels of different deadweight tonnage on a fixed 100-30 building program.

2. Constructing cargo vessels of different tonnages at a fixed fabricating capacity.

It is evident that a fixed building program involves different fabricating capacities and number of men per way for ships of different tonnage, while, with a fixed fabricating capacity, the men per way remain the same but the building program varies. That a proper comparison between these two conditions might be made, it was assumed that the yard was organized and equipped to build 5,000 deadweight-ton ships on a 100-30 building program.

The conclusions to be drawn from a study of these curves show, beyond any question of doubt, that construction costs are dependent to a very large extent on tonnage, and, consequently, that each individual yard should confine itself to vessels that come within its field. Just what the field of any yard might be is determined from the most economical building program at a given man efficiency for ships of different tonnage. For a 7,500 deadweight-ton cargo ship (see Plate 21) the most economical building program is 113-30 at a man efficiency of 35 deadweight tons. Corresponding to this program (see Plate 22), the fabricating capacity should be 100 tons. The number of ways corresponding to the above would be five, consequently a five-way yard with 100 tons fabricating capacity would be in a better position to build a 7,500 deadweight-ton cargo vessel than a yard differently equipped. Conversely, by the same method, the field of a yard can be determined, always keeping in mind the man efficiency. While a standard man efficiency of 45 equivalent deadweight tons has been offered as a result of actual performance it is suggested that, in yards that do not come up to this standard, a tentative one of 35 equivalent deadweight tons be set up to work to. Unless this is done, it is not possible to figure definitely on results to be accomplished nor keep in close touch with day to day performance.

Note.—The term "equivalent deadweight ton" means performance referred to a 7,500 deadweight-ton cargo vessel as a standard for comparison.

DISCUSSION.

The President:—This paper which has been presented by Mr. Churchill is most interesting and should bring forth some pertinent inquiries and discussion. Is there any member who desires to discuss the paper?

Professor H. C. Sadler, *Member of Council*:—Mr. President and gentlemen, I had the pleasure of being associated with Mr. Churchill in the Emergency Fleet Corporation during the last year and a half, and also had the pleasure of discussing with him a good many of the points that he has brought up in this paper. The paper is certainly worthy of the most careful study by all shipbuilders, as certain of the features he brings out are perhaps not altogether realized by most of them.

When it was evident that the shipbuilding capacity of the country had to be enormously increased to take care of submarine sinkings, the first and perhaps most obvious suggestion made was to increase the number of ways in existing shipyards, and also to erect new shipyards. On the face of it, increasing the number of ways in the yards, with, of course, an accompanying increase in the fabricating capacity, would appear to be a logical solution, but Mr. Churchill has brought out another very important fact, and that is that the question of man efficiency may influence the number of ways; and it does not always follow, if you increase the number of ways in a shipyard, you will get an increase in the actual output of tonnage.

The curves on Plates 23 and 24, I think, bring out that point very clearly, particularly Plate 24. If you will analyze the cost per deadweight ton for a fixed fabricating capacity, the question of the number of ways is very important; in other words, the cost per deadweight ton is a good deal less in the 5-way yard than in a 10-way yard, and the two curves cross at a certain deadweight ton, per man, per year at approximately, in this case, of about 40; so that by increasing the number of ways in a yard it by no means follows you will decrease the cost per ton of production.

The figures given on page 50 show where the possible variation comes in—the variation in cost of about $117 and $158 per deadweight ton, depending on whether the yard can use man efficiency of 65 deadweight tons or 25 deadweight tons per man per year.

The reason why I think these figures ought to be studied carefully is that we have in the past two years developed the shipbuilding industry in this country to a position which was undreamed of a few years ago. We are going to come in competition, and probably severe competition, with the world in the next few years, and if the shipbuilding industry is to remain a profitable one in this country it is essential that we should leave no stone unturned to reduce our costs to the absolute minimum; and a paper such as this, I think, gives us food for very careful thought. There are possibilities of reduction of costs indicated in the paper that perhaps were not realized in the past. I think Mr. Churchill is to be congratulated on the very careful analysis he has made of some of these subjects. The matter is put in a form that any shipbuilder can use. Even if his actual figures are not the same as Mr. Churchill's he can apply his own figures and draw his own conclusions, and I think, in that way, probably we can help to meet competition in the future.

I would like to add that the work of Mr. Churchill represents perhaps one part of the work of the Emergency Fleet Corporation which was not realized by the public at large—the study of increasing the production of ships. Mr. Churchill conducted the investigations during the time that the intensive production was on, which led in a good many cases to a very remarkable increase in tonnage turned out in certain shipyards.

The President:—Is there any further discussion, gentlemen? This paper is one which merits very careful consideration. Like all presentations of this character, a great deal depends upon the correctness of the premises, and two of the very important premises that the Chair notes in glancing over the paper are those relating to man efficiency and overhead cost.

I crave indulgence for even mentioning such things, but we all have to deal with them rather seriously.

Perhaps in his concluding remarks Mr. Churchill will throw more light upon the method by which he arrived at the relative efficiency of man power and organization; also some further light upon this "certain fixed overhead expense." These remarks are in no sense to be taken as adverse criticism, but purely as a desire to obtain further light upon a subject which is of the very greatest importance in ordinary times and becomes of extraordinary importance under war conditions.

In the absence of further comment, the Chair requests Mr. Churchill to respond to the remarks that have been made.

Mr. Churchill:—I think the comment by our president is very timely; in other words, the question of how we arrived at the basis of our premise. It is really a very simple matter, after it is told. In the records of the Emergency Fleet Corporation we had weekly and monthly reports of the number of men employed in the different yards, separated as to office force, men engaged on plant construction, men engaged on ship construction for the Emergency Fleet Corporation and men engaged on repair work and other work, such as naval work. In that way we got the records for practically the whole period of the war as to the actual number of men employed on ship construction. That did not mean the number of men actually in the yards, but the total number of men necessary to build ships, exclusive of the office force and men engaged on plant construction and others. Then, with reference to various navy yards, we had records of the output in terms of deadweight tons. This is a very simple matter to arrive at the output per man per year, expressed in deadweight tons.

The President:—In comparing the relative efficiency, as between men and management, you gave the men 20 per cent?

Mr. Churchill:—In order to answer that question it is necessary to go into the matter of the rather peculiar conditions that existed with special reference to new yards. We all know a great many new yards were started by men who were not experienced in shipbuilding. That is well shown when you consider that the average number of men employed in shipyards prior to the war was about 50,000 for all the country, which developed, as a matter of fact, into 360,000 in a little less than one year. As I remarked in reading the paper, it was next to impossible to provide efficient supervision for such a force as that—supervision by skilled men—as it did not exist. As a matter of fact, they robbed the old line yards to a large extent, and that was the best they could do.

In arriving at the estimate of 20 per cent due to the men, and 80 per cent due to management, the main difficulty we found was lack of control. By that I mean the schedules of erection, the method of controlling material, the method of controlling costs were, for the time being anyhow, almost totally lacking in the new yards. That the 20 per cent is assigned to the men is largely a matter of inexperience, judged by the production men in bolting up, driving a certain number of rivets per hour per gang, and other matters in the craft of shipbuilding, and these figures of 80 per cent and 20 per cent are, of course, simply estimates. An analysis could not be made within the short time at our disposal, and it simply indicates that had the ship construction been better controlled through the three methods of which I have spoken—cost, production and material—virtually about 80 per cent of the difficulties encountered in the early stages would not have existed.

The President:—Then you count the efficiency of the men, without management, as 20 per cent, and good management as adding to their efficiency 80 per cent—you do not take into consideration the question of labor turnover?

Mr. Churchill:—That is part of it. For instance, you know, in a great many cases you would have men under instruction in the matter of riveting, and it would take anywhere from two weeks to three months before you could put these men out into the yards to do work in riveting on a piece-work basis and earn a regular day's wage. After a man finally did learn so that he could accomplish a satisfactory output, some other yard got him, and the men were constantly going back and forth.

That is one of the elements to be taken into consideration—labor turnover—that was given due consideration. It was very difficult, due to the method of scamping, as we called it, which existed where the demand for men was so enormous, to prevent men from going from one yard to another; and the way we tried to stop it definitely, as you all know, was through the medium of the Navy Board, and the practical means taken was to equalize wages all over the country, but in the early stages of the game, each yard had scouts out trying to get any workmen who showed a capability to make satisfactory production.

As to the question of overhead, of course, that is a question which is very much thought of, and is a very deep subject; but we found in looking over the cost of the ships—giving the average time of construction, we will say, ten months—the average overhead, based on direct labor employed in the yard, figured out so that it was approximately $1,000 per ship per day. Under normal conditions you will find that is not far off. Of course, the more efficient the yard is the smaller it will be, and the more inefficient the yard is the larger it will be, but all the figures showed approximately, for the purpose of preliminary calculations and comparisons, that $1,000 a ship per day under normal conditions will represent approximately the overhead.

The President:—The difference in the size of ships would be a controlling factor?

Mr. Churchill:—Yes, that would be a factor.

The President:—What do you figure it as a percentage of direct labor?

Mr. Churchill:—It would run anywhere from 60 to 80 per cent of direct labor. There is an accepted rule of thumb that the cost of the ship is represented by a ratio of 2, 2 and 1, 2 for material, 2 for labor and 1 for overhead, or 40 per cent of the cost for material, 40 per cent for labor and 20 per cent overhead. This figures out that the overhead is 50 per cent of the direct labor. As a matter of fact, there are very few yards that kept their overheads down to 50 per cent of direct labor during the war.

The President:—The explanations given by Mr. Churchill will add very much to the value of his paper. The reason that the Chair was specially desirous of knowing something more about the bases of the calculations was that the remark has been repeatedly made to him in his official capacity that the very great labor turnover, or extreme dilution of skilled labor, as well as various other causes which could not be controlled, and did not reflect particularly upon any one, seriously impaired the efficiency of work as compared with normal peace-time conditions. Of course, as Mr. Churchill states, his premise is necessarily an arbitrary one, and one with which perhaps many would disagree, but it gives a basis of comparison.

This paper is one which will bear a great deal of study and profitable study, and I am sure that the members will desire that the Chair express the thanks of the Society to Mr. Churchill for the time and trouble expended in the preparation of this admirable paper.

Before announcing the next paper, the Chair would like to state that there will be a meeting of the Council immediately after the morning session—which will terminate after the reading of the next paper—and I also wish to announce that due to the courtesy of the New York members, the Entertainment Committee, of which Mr. C. M. Wales is chairman, has arranged that the steamer Chester W. Chapin will take the members of the Society and their guests (including ladies) to the works of the Submarine Boat Corporation on Newark Bay to witness a launching. Luncheon will be served on board. The steamer leaves Pier 40, North River, at 11 o'clock on Saturday morning. Tickets may be obtained from Mr. Kain in the rear of the hall.

The next paper on the program is entitled "An Analysis of the Isherwood System of Ship Construction," by Mr. John Flodin, Associate.

Mr. Flodin:—Mr. President and gentlemen, rather than give you a dry reading of a dry paper, or a synopsis of a dry paper, I should like to give you a little sketch of the reasons why I started to do this work. I was very fortunate, when I first started out in shipbuilding, to work with a man of long and varied experience, but like a great many men of long and varied experience, he was very much opposed to the Isherwood system—so much so that he was unwilling to consider it at all. I asked him some questions about it, but he always declined to consider the subject. Finally I started to gather and study magazine articles and advertising literature and various odds and ends, but advertising literature in general is not to be relied on, and while some of the magazine articles I ran across were very interesting—for example, an article by Montgomerie in *Engineering*, for, I think, March, 1913—these articles seemed to take too roseate a view of the subject. Consequently I was glad, indeed, when the shipyard I was then connected with decided to change from the transverse framing construction to the Isherwood type of construction. This gave me an opportunity to compare directly the two systems, but unfortunately it is not possible to give the complete calculations in a paper of this kind—it would take several times the length I have used to give these calculations in full, and consequently many considerations are left out.

One thing I possibly should have mentioned is the fact that in the Isherwood ship all the longitudinals were included in calculating the section modulus, but were reduced the same as the shell plating for riveting—that means, the same deduction was made as for non-watertight rivet spacing as in the shell plating. I mention this, because I anticipate that it will be brought up in the discussion which will follow.

AN ANALYSIS OF THE ISHERWOOD SYSTEM OF SHIP CONSTRUCTION.

By John Flodin, B. S., M. E., Associate.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

Anyone familiar with the history of the art of shipbuilding will admit that the advancement within that art has been very slow. It is no doubt true that this slowness has been largely due to the fact that the growth was dependent on the development of the mechanical arts and the development of land transportation, but it is unquestionably also true that the strong traditions, amounting in many instances to superstitions, that have enveloped all things maritime have formed a serious barrier against all advancement. We have been told many witty and amusing tales of what people thought and said about the first railroad engine and the first trolley car, but only the ignorant are represented in these stories. In the shipbuilding field, on the other hand, we hear of a continuous string of predictions of disasters, not from people who could not be expected to know anything about the matter, but from men whose experience at sea or in shipyards should have enabled them to form something of an opinion of matters of this kind. Yet the prediction that the first iron vessel is "bound to sink" expresses the belief of men who should have known how and why a ship floats. In the same category fall the predictions regarding the Great Eastern, and the expressions of mirth and derision that gave rise to the name "Fulton's Folly."

It is no wonder, then, that any departure from what had become established practice would be frowned upon, and that anyone trying to introduce an improved system of framing would have a hard battle to fight. A very considerable proportion of the shipbuilders, both in this country and elsewhere, regarded with evident doubt the new system devised some thirteen years ago by J. W. Isherwood. While shipowners would have been quite willing to take advantage of the increased deadweight capacity claimed for the longitudinally framed ships, they generally preferred to wait until someone else had taken the risks of experimenting. They wanted to be shown, nor can that be held against them, for who has not been misled by too roseate advertisements? But the new system stood the test of criticism and of many trials and generally gained ground in spite of the insistence of its opponents in characterizing it as a "get-rich-quick scheme," as a freak, and as an imposition on everyone concerned. But have these opponents been entirely wrong? Has the ground been gained on account of the increased deadweight cargo capacity—granting that that claim be true—at the expense of strength and durability?

In other words, are the claims, not only as to increased cargo capacity, but also as to strength, true? It is the first purpose of this paper to endeavor to answer these questions in the case of one specific type of vessel, the choice of which depended on the fortunate circumstance that a western shipbuilding firm, with which I was at the time connected, changed from the transverse to the longitudinal system of construction. Other advantages claimed for the Isherwood system will also receive consideration.

The particulars of the design under consideration are:—Length between perpendiculars, 380 feet; length, over all, 395 feet; length, Lloyd's, 380 feet 3 inches; breadth, moulded, 53 feet; depth, moulded, 29 feet 3 inches; load draught, about 23 feet 6 inches; block coefficient, .790; deadweight cargo capacity, about 7,500 tons.

The two vessels are, as shown by the profiles (Plates 34 and 35), of the poop, long bridge and forecastle type, and both are built to the highest class of Lloyd's Register of Shipping.

In making this change in the system of construction, several minor alterations became necessary or desirable. Thus, for example, the engine and boiler casings and the deck-houses were changed, the fore-peak bulkhead was moved forward, and a bulkhead was added in hold number two, at the forward end of the bridge. But since these alterations are not chargeable to the change in the system of construction, they have either been disregarded or their effect has been equalized.

Notwithstanding the necessity to make these departures from the working drawings, this opportunity for direct comparison offers very considerable advantages. Vessels of both designs have been built and are at the present time in actual operation, so that the case cannot be regarded as a hypothetical one. We have, further, the advantage of being able to obviate reduction to standard conditions or, in the case of steel weights, to a basis of tons of deadweight capacity per ton of steel, since the vessels are of identically the same dimensions.

In this connection attention is called to the absence of side stringers in the transversely framed vessel, this omission having been compensated for by increased stiffness of side framing and greater thickness of shell plating. Since this simplification is now quite commonly adopted, both ships may be regarded as typical of the system of construction that they represent, but it should be borne in mind that the transverse framing construction has gone through a long course of development, while the Isherwood system is relatively very young and may consequently be regarded as being in a rather rapid state of evolution.

COMPARISON OF WEIGHTS AND CAPACITIES.

Rather careful calculations of the net steel necessary to complete the two vessels show that 2,023 tons were used for the transversely framed vessel and 1,881 tons for the Isherwood ship. These figures, which are in tons of 2,240 pounds, do not include scrap; that is, deductions have been made for lightening holes, notches in the plates where continuous members pass through, etc., and no allowance has been made for rivet heads or liners. The last two items should be very nearly the same for both types of ships, so that our comparison will not suffer from the omission of them.

The saving in net steel amounts, then, to 142 tons, or slightly over 7 per cent, certainly a saving well worth while provided it is not accompanied by a reduction in strength. This, however, does not represent the increase in cargo-carrying capacity, for the reduction in weight on the basis of the completed ship, including all machinery, equipment, etc., equals only about 4.65 per cent, and the increase in the deadweight carrying capacity is about 1.9 per cent. To the shipowner this last figure is the only one of any great significance, for since it may be assumed that the custom-house measurements—that is to say, the taxes levied upon the vessel—are not affected by the change in the system of construction, the increase in the deadweight capacity equals the increase in the earning capacity in all cases where the weight, rather than the volume, is the important factor, unless, of course, the cost of operating the Isherwood vessel should rise because of the need of more frequent or more costly repairs.

The increase in the earning capacity is, in fact, the principal claim for favor made by the supporters of the Isherwood system, coupled, as it is, with the claim that the grain and bale capacities are also increased. The former contention is undoubtedly true, the improvement being due partly to the fact that the saving in steel means that a smaller volume is occupied by the framing, but mainly to the greater facility with which the inner bottom ceiling may be run out horizontally to the shell plating of the Isherwood ship, instead of being run up along the faces of the margin brackets, as is commonly done in transversely framed vessels. The bale capacity (*i. e.*, the actual, "as loaded" bale capacity, not the calculated), on the other hand, is naturally somewhat impaired because the transverses break into the cargo holds at intervals that are but rarely a multiple of any of the three dimensions of the bales. In case of ordinary bale goods this may not be a serious matter, but in case an Isherwood ship were to carry lumber it would be necessary to load "short stuff" (that is, lath, shingles, or bundled flooring or ceiling), which means a mixed cargo where a straight cargo might be more easily obtainable, not to mention the greater stevedoring charges.

In this connection attention is called to the fact that practically all oil tankers are now being built to the Isherwood system, the principal reason being that this system lends itself more readily to the peculiar bulkhead and stiffening work necessary in that type of vessel, and only secondarily to the saving in steel weight. The increase in the cubic capacity has, of course, no influence on the choice of system of framing for a tanker, since the deadweight capacity, not the cubic capacity, is the significant factor.

LONGITUDINAL BENDING STRENGTH.

As a criterion of the longitudinal bending strength, we might simply find and compare the section moduli of the two vessels. The matter was, however, gone into somewhat more deeply; stress diagrams for both the hogging and sagging conditions were prepared. These diagrams were worked out to represent stresses that would actually be likely to exist, rather than to conform to the so-called standard conditions. The result was that the stresses, especially for the sagging condition, are somewhat lower than those developed under standard conditions for vessels of this size, the bending moments being 41,200 foot-tons for hogging and 9,800 foot-tons for sagging.

The bending moment is maximum at about the middle of length for both the hogging and sagging conditions. But at this point the ship is strengthened by the long bridge, so that the point of most unfavorable combination of bending and resisting moments falls either just forward of the forward end of the bridge, or just abaft the after end of the bridge, at which points the section moduli may be taken as being equal since the scantlings are the same at both points, only the sheer heights changing slightly. The section moduli were, in fact, calculated for the lowest point of sheer, and they may consequently be regarded as being somewhat conservative. On the other hand, in working out these section moduli an important departure from standard practice was made: the main rail and bulwark plating were included as strength members. The reason for this was that in the Isherwood ship the main rail and one of the bridge-space shell longitudinals form a practically continuous member. Since the shell longitudinal certainly is a strength member, there is good reason for including the main rail, and consequently also the bulwark plating, in the section modulus of the Isherwood ship, especially when these members are in tension. And if we decide in favor of the inclusion in the case of the Isherwood ship, the corresponding members must, of course, also be included in the section moduli of the transversely framed vessel, for the sake of retaining a fair basis of comparison.

The values of the section moduli, in inches to the third power, are:—

	Hogging condition.	*Sagging condition.*
Transversely framed vessel..................	161,900	178,100
Isherwood vessel	170,300	189,700

In each case a deduction for seven diameter rivet spacing was made from the sectional area of the tension part of the section.

By applying the flexure formula, the unit stresses, in pounds per square inch, are now found to be:—

	Hogging condition.	*Sagging condition.*
Transversely framed vessel..................	6,840	1,477
Isherwood vessel	6,500	1,392

It will be seen that the fiber stresses are, roughly, 5 per cent less in the Isherwood vessel than in the transversely framed vessel.

This comparison holds good only, of course, when the vessel is in the upright position and the plane of bending is vertical, but it seems safe to conclude that an approximately similar relation of strength would be shown to exist were we to continue the analysis for various angles of inclination, and we may make the general deduction that in longitudinal bending strength an Isherwood ship is materially superior to a transversely framed vessel of the same dimensions.

VERTICAL SHEARING STRENGTH.

A similar inference may be made for the vertical shearing strength. From the stress diagram for the hogging condition it was figured that the maximum vertical shearing stress is 466 tons, which is resisted by a cross-sectional area of 2,004.5 square inches for the transverse vessel and 2,252.7 square inches for the Isherwood vessel, giving unit shearing stresses of 520 and 468 pounds per square inch, respectively. While there is a material reduction in the unit stress in favor of the longitudinally framed vessel, this can scarcely be credited as an advantage for the Isherwood system inasmuch as the stress values are too low for consideration.

LONGITUDINAL SHEARING STRENGTH.

Longitudinal or horizontal shear is usually found by the formula:—

$$\text{Longitudinal shear} = \frac{VM_s}{tI}$$

where V = vertical shear.

M_s = the statical moment of the section (= area $\times$ distance) between the plane at which it is desired to find the longitudinal shear and the extreme fiber, on one side of the plane of shear.

t = thickness of web at plane of shear,

I = the moment of inertia of the section.

In applying this formula to ordinary structural work, no consideration is given to transverse stiffeners or brackets, and consequently no careful investigation seems to have been made in this field. It is evident, however, that if the hull were to fail because of longitudinal shearing stresses, say along an edge lap in the shell at or near the neutral axis, the transverse framing would have to be sheared off along that edge lap. In the transversely framed ship we have transverse channel frames spaced 2 feet 2 inches, continuous from the bilge to the upper deck. These frames have a sectional area of 8 square inches each, but because of the elasticity of the riveting connecting the frames to the shell plating, it would probably be unsafe to regard more than 50 per cent of this area as effective in longitudinal shear.

At the edge lap between H and J strakes we have, then, per frame space, for one side of the ship:—

	Square inches.
Gross shell plate area = 28 x .66	18.48
Less rivet holes, which for 4½ diameter spacing = 2/9 gross area.	4.10
Net plate area	14.38
Effective frame area	4.00
Total area in shear	18.38

This corresponds to 8.47 square inches per foot length of ship for one side, or 16.94 square inches for both sides of the hull. The equivalent web thickness would then be 1.41 square inches.

In the Isherwood ship the case is quite different, however. We have, as before, the area of the shell plate, which amounts to 5.6 square inches per foot of ship, for one side, after deducting for the riveting. But to this area we cannot add the strength of a transverse frame. The angle connecting the transverse web frame to the shell is cut at every longitudinal, and the web plate itself is deeply notched, giving sufficient elasticity in the transverse members to make them useless in longitudinal shear. It is true that some frame rivets between the edge lap and the longitudinal would have to be sheared. If, in our case, we take the shell lap between *H* and *G* strakes as the plane of longitudinal shear, four or six rivets connecting the transverse clip to the shell between that lap and longitudinal number 16 would add to the longitudinal shearing strength. But with transverses spaced 12 feet apart, this becomes a negligible quantity and we can use the value found above, which gives an equivalent web thickness of 0.932 inch as the net steel effective in longitudinal shearing.

Using these values for *t* in the formula given above, 466 tons as the vertical or transverse shear and the proper respective values for the static moments and moments of inertia, we find that the longitudinal shear for the transverse vessel amounts to 2,570 pounds per square inch, and for the Isherwood vessel to 3,920 pounds per square inch, which shows an increase in fiber stress of over 50 per cent. In either case the stress is small, however, and the reduction in strength should not be regarded as endangering the safety of the Isherwood vessel.

TRANSVERSE STRENGTH.

Although not as important as the longitudinal stresses, the transverse stresses a vessel is subjected to are often quite severe, and it is consequently essential that a change in the system of construction does not unduly weaken the vessel transversely any more than a weakening longitudinally would be permitted. For example, if data based on long experience seem to indicate that a certain bilge connection furnishes but a reasonable margin of strength, we cannot arbitrarily substitute another connection of less strength without adequately compensating for the loss. By doing so we should expose the vessel to serious straining when, especially if heavily loaded, it encounters a seaway. Immediate failure would perhaps not result, but the riveting would gradually give away, and troublesome leaks would develop.

Failures due to the three principal transverse stresses are indicated in Figs. 1 to 5, inclusive (Plates 36 and 37).*

DIAGONAL DISTORTION.

A distortion of the hull in the manner and because of the stresses indicated in Figs. 1 and 2 (Plate 36) would evidently mean a failure of the transverse framing in the vicinity of the bilge, at or near the connection of the second deck to the side of the ship, and possibly, but not necessarily, at or near the connection of the upper deck to the side of the ship. Considering these in the order named, first for the transverse vessel and then for the Isherwood vessel, we have:—

The Bilge Connection.—From an inspection of the midship section we can conclude that the weakest point in the bilge connection is either through the riveting connecting the side frame to the bilge bracket, or through the frame itself, in line with two of the connecting rivets. The failure of the framing would in either case be accompanied by a failure of the shell plating, so that we should include, as contributing to the strength of the framing, a strip of the shell plating equal in width (*i. e.*, in the longitudinal direction) to two and one-half times the width of the shell flange of the frame bar, reduced by one-half the diameter of the shell rivet in order to obtain a result that will represent a mean between tension and compression conditions.†

To determine the strength of the riveted connection the polar moment of inertia about the center of strength of the section must be found. This is most conveniently done by first finding the center of gravity of the connection and then calculating the moments of inertia about vertical and horizontal axes through the center of gravity. The sum of these two moments of inertia equals the desired value of the polar moment of inertia.

Taking the shearing strength of the rivets as equal to the tensile and compressive strength of the steel, the moment of inertia about the horizontal axis is found to be 577.5 inches^4, and about the vertical axis 157.5 inches^4, from which the polar moment of inertia equals 735 inches^4. If we regard the distance of the center of the farthest rivet from the center of gravity of the connection (or, more correctly,

*These sketches are borrowed from Holmes' Practical Shipbuilding.

†The width of the strip of shell plating that may be regarded as contributing, to the full extent of the tensile strength of the material, to the strength of the frame bar or stiffener is variously given as a function of the thickness of the plating and as a function of the width of the connecting flange. The conservative values here used, namely, two and one-half times the flange width for single bars and one and one-half times the sum of the flange widths for double bars, do an injustice to the Isherwood ship, inasmuch as the width of the shell strip included for the transversely framed vessel is 8.25 inches for each frame space, or 31.7 per cent of the total shell; while for the double angle connection of the Isherwood vessel the width of the shell strip becomes 16.25 inches for each transverse, or about 11.3 per cent of the total shell. It is difficult to conceive, however, of a mode of reasoning that would warrant considering equal proportions of the shell as contributing to the strength of the framing.

the center of strength) as the distance to the extreme fiber, the polar section modulus equals $735 \div 13.13 = 55.98$ inches^3.

The moment of inertia of the side frame, deducting for the rivet holes, equals 186.86 inches^4, and the section modulus equals 25.5 inches^3.

The Second Deck Connection.—Four possible modes of failure present themselves at the juncture of the second deck beam to the frame. The beam bracket may be ruptured, as shown in Fig. 1, Plate 36; the frame may fail at the toe of the bracket; the riveting of the bracket to the frame or to the beam may fail; or, finally, the beam may fail at the inboard toe of the bracket. It is very evident, however, that the bracket plate is considerably stronger than the rivets connecting it to the beam or to the frame, so that the first mentioned possibility may at once be discarded. Furthermore, the section modulus of the side frame, without correcting for the riveting or for the strip of shell plating contributing to the strength of the frame, is 21.9 inches^3, as compared to the section modulus of 12.7 inches^3 for the deck beam channel, so that the possibility of failure of the side frame at the toe of the beam bracket may also be abandoned.

For the strength of the bracket riveting, considering the rivets in one arm of the bracket only, we may divide the sum of the second moments of the rivets about the center of the rivet nearest the heel by the distance to the toe rivet. The section modulus thus obtained is 26.02 inches^3. Calculating the strength of the deck beam at the toe of the bracket, including again a strip of plating, we find that the section modulus is 16.1 inches^3.

The Upper Deck Connection.—Applying the same reasoning to the upper deck connection, we find the section moduli for the riveting and for the deck beam at the toe of the bracket to be 18.61 inches^3 and 13.2 inches^3 respectively.

The Combined Strength.—As a representation of the ability of the hull to resist forces tending to cause diagonal distortion, we may, then, take the sum of the least section moduli at each connection, or, excluding the upper deck, $(25.5 + 16.1)$, $\times 2 = 83.2$ inches^3, and including the upper deck, $83.2 + 2 \times 13.2 = 109.6$ inches^3, the factor of two being introduced by the fact that failure cannot occur at one side of the vessel alone.

The corresponding values per foot length of the vessel are 38.4 inches^3 and 50.5 inches^3, the frame spacing being 2 feet 2 inches between peak bulkheads.

Diagonal Strength of the Isherwood Ship.—The corresponding probable points of failure in the Isherwood vessel are:—

At the bilge connection:	At longitudinal No. 14.
At the second deck:	At the second longitudinal from the side of the vessel.
At the upper deck:	At the second longitudinal from the side of the vessel.

These points of failure, while they are taken at the points of least section moduli, are somewhat further removed from the theoretical points of failure than were the corresponding points in the transversely framed vessel, thus introducing the factor of a lever arm. It is impossible to express this difference mathematically, if, indeed, such an expression should be desired, and consequently the difference has been omitted in these calculations. It should be remembered, however, that this omission makes the exactness of the conclusions somewhat doubtful, the doubt being in favor of the Isherwood ship.

Calculating the strength at the points indicated above, using the same reasoning as for the transversely framed vessel, but including as part of the transverse at longitudinal No. 14 a strip of the shell plating equal in width to only one and one-half times the combined width of the shell clip flanges, since a double angle connection is here used, we have, for the bilge connection, a section modulus of 241.0 inches^3; for the second deck connection, a section modulus of 41.7 inches^3; or a total of 565.4 inches^3 for both sides.

The modulus for the upper deck connection is 47.5 inches^3, giving a total of 660.4 inches^3 for both sides of the entire section. The transverse spacing being 12 feet (except in the peaks and forward of the three-fifths length), the combined section modulus per foot length of the ship becomes 47.1 inches^3 excluding the upper deck, and 55 inches^3 including the upper deck.

Summary.—For more ready comparison we may summarize these section moduli as follows, the values being given in inches^3 for both sides of the vessel:—

Connection	Transverse framing		Longitudinal framing	
	Per frame space	Per foot of ship	Per frame space	Per foot of ship
At the bilge................	51.00	23.55	482.0	40.17
At the second deck..........	32.20	14.85	83.4	6.95
At the upper deck	26.40	12.20	95.0	7.92
Totals.................	109.60	50.60	660.4	55.04

It will be noted that the Isherwood vessel compares very favorably with the transversely framed ship as far as the total resistance to diagonal distortion is concerned. The Isherwood bilge connection develops 70.6 per cent more strength than the corresponding connection in the older system of framing, while the Isherwood deck connections are weaker. And it may be argued that this is a more favorable distribution of strength, since the bilge connection is continuously subjected to severe stresses due to the water head. There is unquestionably some justification for this viewpoint, provided that the deck connections are not weakened to such an extent as to make the decks themselves unsafe. This question will, however, be considered separately under the head of "Strength of Decks."

FAILURE OF BOTTOM OF SHIP THROUGH COLLAPSING.

In the case of a failure of the type shown by Fig. 3, Plate 37, which might occur when the weight of the vessel is resting on keel blocks along the center line, or on the ground, as in the case of stranding, a considerable discrepancy exists between the transversely and longitudinally framed vessels. In the former we have two rows of pillars, giving a transverse span of 21 feet between the pillars, while in the latter case there is but a single row of pillars along the center line of the vessel. In the former vessel the stressing of the bottom to failure through collapsing would, then, mean the failure of the floors at or near the center line, while in the latter the most dangerous point of loading would be at some point about halfway between the center line and the support at the margin, and the failure would then occur near the point of loading.

For the transversely framed vessel we find the section modulus to be 243.8 inches^3 for the floor connection at the center line (the line of fracture is supposed to pass through the riveting of the floor clips to the center vertical keel, past the top and bottom horizontal keel angles, and through the flat keel and rider plates) and 283.3 inches^3 for the floor through the first lightening hole from the center line of the ship. In each case a strip of the shell plating equal to two and one-half times the horizontal flange of the connecting angle is included, and also a strip of the tank top plating of similar width less one rivet diameter, the tank top plating being in tension, while the shell plating is in compression.

The least section modulus is, then, 112.2 inches^3 per foot length of the ship. Regarding the floor as an encastre beam,* and taking the allowable fiber stress as 16,000 pounds per square inch, we may find the safe concentrated load by the formula—

$$M = \frac{Pl}{8},$$

where M is the bending moment in inch-pounds, P is the load in pounds, and l is the length of the span in inches.

Equating this to the resisting moment we have—

$$P = \frac{16{,}000 \times 112.2 \times 8}{21 \times 12} = 57{,}000 \text{ pounds per foot of ship.}$$

The area of a floor section through a lightening hole near the pillar is 22.69 square inches, or about 10.5 square inches per foot of ship, giving a maximum shearing stress of $\frac{1}{2} \times 57{,}000/10.5$, or about 3,000 pounds per square inch.

In the Isherwood ship, the failure would undoubtedly occur through a lighten-

*It would be more reasonable to consider the floor a continuous beam, the assumption here made not being strictly correct, but since this error of conception will be used for the Isherwood ship also, the final result will be a very close approximation to the true relation in the strength of the two vessels.

ing hole, the line of fracture passing through the notches for both the tank top and bottom shell longitudinals. The modulusfor this section is 303.2 inches^3 for each transverse floor and for the intermediate floors, or 50.5 inches^3 per foot of ship.

Regarding the floor again as an encastre beam, whose span in this case is 20 feet (allowing a reasonable overlap with the side transverse bracket), we have—

$$P = \frac{16{,}000 \times 50.5 \times 8}{20 \times 12} = 26{,}900 \text{ pounds per foot of ship.}$$

It will be seen that the strength of the Isherwood ship falls far short of the strength of the transversely framed one as far as the collapsing of the bottom is concerned. It should be noted, however, that the center-line arrangement of the pillars in the Isherwood ship gives that ship an advantage when resting on keel blocks. But this pillar arrangement is not a characteristic of Isherwood ships in general, since many of them have two rows of pillars, and would then show a weakness as compared with the transverse construction that is roughly represented by the results obtained above. In either case this strength represents resistance against extraordinary external forces and is not to be regarded as representing conditions met with in ordinary service. The stress on the bottom of the ship, using the load draught of 23 feet 6 inches, but not making any allowance for the balancing effect of the cargo, would set up a fiber stress of about 5,800 pounds per square inch in the floor of the Isherwood ship, showing that for sea conditions the Isherwood construction furnishes a large margin of strength.

It should further be remarked that Lloyd's rules permit the omission, in transversely framed vessels, of alternate floor plates, except forward of the three-fifths length and under the engines and boilers, at which parts Isherwood vessels have additional floors. In cases where alternate floor plates are omitted, only the bottom frame bars and the reverse frames are fitted, and even the reverse frames may, under certain conditions, be omitted. A vessel so constructed would show very nearly the same strength of bottom as the Isherwood vessel, namely, a maximum safe load of 28,500 pounds and 26,900 pounds for the two types respectively per foot length of ship.

In view of the above figures it may, then, be said that even though the Isherwood vessel does not possess the same strength of bottom, per foot length of ship, as a transversely framed vessel of the same size and type, built with floors at every frame, the strength developed by the Isherwood construction is ample, and the construction is consequently more logical.

THE SHELL PLATING.

On the midship sections (Plates 32 and 33) the shell plating is seen to be lighter for the Isherwood vessel. The fact is, however, that the thickness of the shell plating of the Isherwood vessel is in accordance with Lloyd's rules, regardless of the change in the system of construction, while the shell plating of the transversely framed vessel was increased in thickness to compensate for the omission of the side stringers. Hence no comparison of the strength of the shells need be made, except in so far as they enter into the longitudinal strength of the ships.

COLLAPSING OF THE SIDE FRAMING.

In connection with the analysis of the resistance of the two vessels to diagonal distortion, it was shown that the strength of the Isherwood transverses at longitudinal No. 14 is represented by 40.17 per foot of length of vessel, as compared to 23.55 for the strength of the side frame of the transversely framed vessel at the top of the bilge bracket. The Isherwood transverse at the point considered is about one inch deeper than it is at the longitudinal next above, so that the strength ratio of 70.6 per cent, in favor of the Isherwood vessel, does not quite apply to the strength of the side framing. But owing to the great excess in strength of the Isherwood transverse, we need have no hesitancy in saying that there is a considerable margin of strength of side framing in favor of the longitudinally framed ship—a margin of, perhaps, about 68 per cent.

STRENGTH OF DECKS.

As in the case of the strength of the floors, the difference in the pillar arrangement introduces a considerable discrepancy in the deck strength of the transversely and longitudinally framed vessels. In the transversely framed vessel the strength of the decks is obviously dependent on the strength of the deck beams and on the maximum span, which is again the distance between the pillars, or 21 feet. Including, as before, a strip of the deck plating as contributing to the strength of the beam, the section moduli of the deck beams are found to be 12.61 inches^3 for the upper, and 15.2 inches^3 for the second deck beam, or, per foot of ship, 5.81 inches^3 and 7.01 inches^3, respectively. Using a fiber stress of 16,000 pounds per square inch and applying the formula for uniformly loaded encastre beams, the upper deck beams are found capable of supporting 8,860 pounds per foot length of ship, or 423 pounds per square foot of deck area (a rather high value, since it corresponds to a 6.75-foot head of water—more than twice the height of the bulwarks—or a 10-foot deck load of lumber); while the safe load for the second deck is 10,700 pounds per foot length of ship, or about 510 pounds per square foot of deck area (again a rather excessive value, since it would allow the carrying of a homogeneous cargo weighing 64.5 pounds per cubic foot, a figure considerably above what the buoyancy of the vessel could support).

It will be noted that the weakest part of the decks of the transversely framed vessel is between the pillars, that is, between the hatches. In the Isherwood ship the load on the decks between the hatches is supported by the deck longitudinals, from which the stresses are transmitted to the bulkhead at the middle of the longitudinal span between the hatches and to the hatch end transverses at the ends of that span.

Calculations similar to those made for the decks of the transversely framed vessel show that the deck longitudinals are capable of supporting loads of 2,290 pounds and 3,310 pounds per foot length, for the upper and second decks respectively. Because of the variations in the spacing of the longitudinals, this means that the maximum safe load for the upper deck is 665 pounds per square foot between the hatches and 723 pounds per square foot between the hatch-end transverses outboard of the fore-and-aft hatch coamings, while the second deck could carry a load of 945 pounds per square foot between hatches and 830 pounds per square foot in the wings.

This, however, does not take care of the strength of the decks at the wings along the sides of the hatches. Here the loads on the longitudinals are transmitted to the half-transverses that furnish the intermediate reactions. These half-transverses have section moduli of 36.55 inches^3 and 34.3 inches^3 for the upper and second decks, respectively. Assuming that the decks are uniformly loaded up to the smaller maximum capacity found above for the space between the hatch-end transverses, and that the load is concentrated at the points of support of the longitudinals, we can proceed to determine the fiber stresses. In either deck the extreme outboard longitudinal is directly supported by the transverse deck bracket, so that the loads borne by that longitudinal and directly by the shell connection do not add to the bending moment. For the rest of the transverse beam the bending moments are approximately 500,000 and 520,000 inch-pounds, giving fiber stresses of 13,600 and 15,200 pounds per square inch in the upper and second deck half-transverses respectively.

The inboard ends of these half-transverses are supported by the fore-and-aft hatch coamings, which in the Isherwood ship are somewhat stronger than in the transverse vessel. The reactions of the fore-and-aft hatch coamings are in turn borne by the hatch end transverses, which also receive the reactions from the ends of the longitudinals between hatches and outboard of the fore-and-aft coamings, the last loads being similar to those borne by the half transverses. These hatch-end transverses are heavily built, and are reinforced by a 10-inch by 12-inch face plate for the second deck, and a 10-inch by 0.70-inch face plate for the upper deck, although these are not shown on the midship section (Plate 33). The loads are heavy, however, and the fiber stress at either deck is somewhat in excess of 40,000 pounds per square inch. Since the bending moment, and hence the fiber stress, are directly proportional to the loads, we may conclude that the safe loads found above are about two and one-half times too great. The deck loads for the Isherwood vessel should, then, be about 280 pounds per square foot for the upper deck and 330 pounds per square foot for the second deck, as compared to the corresponding values for the transversely framed vessel of 423 pounds and 510 pounds per square foot.

Apparently, then, there is a loss in the strength of the decks amounting to nearly 34 per cent for the upper deck and 35 per cent for the second deck.

It was pointed out, in connection with the strength of the decks of the transversely framed vessel, that a large margin of strength existed. Some reduction could consequently be approved of, but it seems that it would be more advisable to increase the strength of the hatch-end transverses and reduce the surplus strength of the deck longitudinals, especially so since the fiber stresses due to longitudinal bending of the ship as a unit are unduly small.

This is, however, a question pertaining to the particular design under consideration rather than to the principles of the Isherwood system of construction.

MISCELLANEOUS STRESSES.

Torsion.—The question of torsional strength would probably have escaped attention had it not been for the fact that the inquiries, which will be referred to later, elicited the information from a European shipping firm, who request that their name be not used in connection with the statements they make, to the effect that the two Isherwood ships they own and operate showed a torsional weakness. To quote their letter, in free translation:—"The ships seemed to be looser, allowing an undue amount of twisting. This made it necessary to stiffen them by means of additional bracketing, which successfully overcame the difficulty, but which also partly wiped out the saving of weight above referred to. But, as already mentioned, these were some of the first longitudinally framed ships ever built, and we understand that the later vessels of this kind are proving very satisfactory. * * *"

Since the torsional strength of any body is proportional to the polar section modulus, and since it is evident that the increased number and areas of the longitudinal members in the Isherwood ship mean an increase in the polar section modulus (*cf.* the section moduli for horizontal axes), it is rather difficult to understand how an Isherwood ship can allow "an undue amount of twisting." The only explanation that presents itself is that the shell and deck longitudinals were not properly connected where they were cut at the watertight bulkheads, a suggestion that appears to be a probability when it is remembered that additional bracketing successfully overcame the difficulty. But even if the bracketing were originally faulty, the shell and deck plating should have remained intact, giving these vessels practically the same polar section modulus as they would have had if they had been built to the transverse system, and they should consequently not have shown any torsional weakness.

It is not impossible, however, that the question of torsion is not fully understood; it may be that the transverse stiffening of a hollow body has an effect on the torsional strength that is not represented in the polar section modulus, but it seems probable that this effect, if it does exist, would be slight. It appears, then, very unlikely that a properly designed and constructed Isherwood ship should be weaker in torsion than a transversely framed vessel of the same type and dimensions.

Stresses Due to Vibration.—It is contended, and is probably true, that vibration is less in Isherwood ships than in transversely framed ships. But it should be remembered that the question of vibration becomes serious only when there is synchronism between the vibration period of the hull, the vibration period of the machinery, or the vibration period of liquids in tanks or liquid cargo, or between all three periods. Enough work has not been done in this field to make it possible to predetermine any one of these three vibration periods, except in so far as reciprocating machinery may be regarded to be likely to vibrate in synchronism with the period of the stroke, and it consequently seems decidedly advantageous to reduce the amplitude of the vibrations wherever that is possible.

PRACTICAL CONSIDERATIONS AND OPINIONS OF SHIPBUILDERS AND SHIPOWNERS.

Among the advantages claimed by the Isherwood supporters are that the ships built to this system are more easily cleaned and repaired, and that the ventilation is improved.

It is unquestionably true that the Isherwood ships are more easily cleaned and painted, especialy so in the double bottom. With the floors spaced 24 to 30 inches apart, as is usually the case in transversely framed ships, the cleaning and painting of the spaces between the floors become both an expensive and unpleasant task, whereas with the floors spaced from 5 to 6 feet apart, as is the practice in Isherwood ships abaft the three-fifths length, there is ample space and better air for the men to work in.

It is not so easy, however, to draw a conclusion as to the ease of repairing the vessel. This is a point on which the results of years of experience alone can throw light. This is also true of the question of ventilation.

In order to obtain information on these points, as well as general expressions of opinions and criticisms, inquiries were sent out both to shipowners who were or had been operating Isherwood ships and to shipbuilders who had had extensive experience in this field of work. In all forty-eight letters of inquiry were sent out, the field covered including, besides American firms, several European countries, Australia, and China. The results were, unfortunately, not very satisfactory. Many of the inquiries, or the letters sent in answer to them, were perhaps lost or miscarried because of the war-time irregularities in the mail service, and of the relatively few answers that were received a considerable number were non-committal, or the writer had the impression that the Isherwood system had certain advantages but admitted that he did not know this from his own experience. With but one exception (see reference to letter from a European firm in paragraph on torsion), all the answers received were favorable to the Isherwood system, and some of them contained statements of sufficient value and definiteness to be worthy of quoting here:—

From W. R. Grace & Co., New York City:—

"The Isherwood system has advantages and disadvantages compared with the ordinary system of framing. The advantages are, that in way of the cargo hatchways the Isherwood transverses form an arched system of construction, giving better support to the deck and dispensing with girders and, to a great extent, pillars. One disadvantage is that the transverses take up more depth than the ordinary frames, and cargo has got to be stowed between the transverses. Against this is the fact that webbed frames can be dispensed with.

"In repair work, we consider the Isherwood ship is as easily repaired as an ordinary frame ship. The longitudinals on an Isherwood ship run practically the whole length of the hold, but in case of damage to the shell or bottom, we have in all cases allowed the repairers to cut these longitudinals and connect them with a bosom piece and strap and find that this does not in any way impair the strength of the vessel. This obviates the necessity of removing a great deal of the shell or bottom plating in order to get out the longitudinals for full length.

"I do not think that I can add to the above, but consider that from both a technical and practical point of view, the Isherwood ship compares very favorably (more especially for ships carrying oil in bulk) with vessels built to the ordinary system of framing. * * *"

From an English shipping firm, who request that their name be withheld:—

"* * * Both of these ships have now been at sea for over five years, and they have so far proven quite satisfactory in every way. They are reported to us as being easy in a seaway, and, owing to the longitudinal system on which they are built, the ventilation is good. * * * The double bottom is easily cleaned and scaled, and the holds are readily accessible for cleaning. There is an absence of vibration, and we have had no signs of straining in upper structures. * * *"

From Lawther, Latta & Co., London, England:—

"* * * Broadly speaking, the Isherwood system of longitudinally building ships appealed to us as allowing a much cleaner job. Experience has further verified that there is less liability to bad workmanship, as it is easier for a workman to make a good job than to scamp his work. For steamers of increasing length, and when resting on two waves, the center part of the vessel may be said to be not altogether fully water-borne, so that the Isherwood longitudinal strength in this department is undoubtedly advantageous. We have found no weakness in front of the bridge, which was quite common in the other type of vessel. There is undoubtedly less vibration, and in consequence we imagine the general upkeep of Isherwood steamers will prove to be less expensive, and in the absence of something intervening in the meantime which we have not so far detected, we should imagine as old vessels they will prove more efficient, as being more free from vibration at the crucial time than vessels built on any other principle. * * * In the case of our S. S. Anglo Brazilian, which had very serious bottom damage at Montreal, we had her repaired at Newport News and were astonished at the ease with which the Isherwood principle of bottom build allowed of such a repair being carried out. Practically the whole of her bottom had to be dealt with, and the complete repair was carried out in the exceptionally short time of nineteen running days. We are not of the opinion that a steamer with the same amount of damage, built on the ordinary system, could have been repaired in anything like the same period. * * *"

From A. H. Bull & Co., New York City:—

"* * * Our steamer Millinocket is the first boat built under this system in American yards. She is now seven years old (letter dated June 7, 1917), so that sufficient time has elapsed for any structural weakness or defects to develop that might show an increase in the cost of upkeep over ordinary construction. We have found no difference in the cost of upkeep. * * *"

From A. F. Kalveness & Co., Christiania, Norway:—

[*Translation.*]

"* * * we have had but one ship built to the Isherwood system, namely, S. S. Storstad.

"This vessel has been employed chiefly in the ore trade betwen Newfoundland and Sidney, C. B., and partly in carrying general cargo. We have found the Isherwood system satisfactory in that the vessel has been strong and has fulfilled our expectations."

The answers received from the naval architects connected with shipbuilding firms were uniformly in favor of the Isherwood system of construction, but it is not impossible that this favorable opinion was to some extent due to the greater ease and economy of fabrication and erection of the Isherwood ship.

The longitudinally framed vessel contains more bracket and small fitting work than the transversely framed one, but this item is more than offset by the lesser amount of furnacing of shapes. In erecting, the Isherwood ship offers two distinct advantages that tend to increase both the speed and economy of construction. One of these has already been mentioned in connection with cleaning and painting, namely, the greater accessibility to the various parts of the vessel, an item that seems even more important in erecting than in cleaning the ship. The second advantage is that the Isherwood ship can be erected in stories, as it were. That is to say, after the inner bottom has been completed, the side transverses up to the lowest deck may be erected, together with the transverses for that deck. All the material then taken on board, such as the deck longitudinals and plating, need be hoisted only high enough to clear the deck reached, not, as is the case with transversely framed vessels, high enough to clear the tops of the side frames, which, for the lowest deck, may mean a difference in the hoist of from 8 to 32—or more—feet, depending on the number of decks. This holds true for all decks except the highest one, although the height of the unnecessary hoist is, of course, reduced as the higher decks are reached.

To show what this apparently small item means to the shipbuilder, let us give a moment's thought to, let us say, a passenger ship of 20,000 tons displacement having four complete decks, the deck height being 8 feet in each case. This means a total height of 24 feet from the lowest to the highest deck, or an average hoist of 12 feet above the lowest deck level for all material for that deck and for the decks above. · A total weight of perhaps about 1,000 tons would, then, have to be hoisted to an average greater height of 12 feet for the transversely framed vessel than would be necessary for the Isherwood ship, a difference of 12,000 foot-tons. That this matter did not receive attention in the comments of shipbuilders is probably due to the circumstance that the power costs are customarily charged to overhead and not to the ship directly, thus obscuring this part of the information.

CONCLUSION.

While the particular ship here used for purposes of analysis and comparison is not above criticism, it should be evident from the above that the Isherwood system of longitudinal framing offers many and important advantages, while the disadvantages seem to be relatively slight. Whether further development in shipbuilding will be along the lines of longitudinal or transverse framing, or a compromise between the two, is a question that the future alone can answer.

DISCUSSION.

The President:—The paper which has just been presented to you, "An Analysis of the Isherwood System of Ship Construction," is now open for discussion.

Mr. F. M. Hiatt, *Member*:—Mr. Flodin, in his opening paragraph, implied that the engineering profession engaged in the art of shipbuilding has been perhaps ultra-conservative.

The problem before the naval architect, however, is not to demonstrate his own daring but to design ships which will enlist the financial support of the investing public. I recently asked a banker to state for me those qualities of any paper which are most essential to insure its successful flotation. His answer was:—"There is only one such quality and that is the security of the invested principal."

The bridge designer can demonstrate the efficacy of his design, including such innovations as he may desire to embody therein, by a test load upon the completed structure. The naval architect, however, cannot at will produce a standard wave for the test of his wares. He must depend upon the test of time. Until that test is passed, the burden of proof is placed upon the designer, not by his conservative colleagues but by the investing public. Nor can this requirement of the investor be unduly criticized. It may be remembered that financially the Great Eastern instanced in the paper was a failure.

It seems to me that the claim of superior longitudinal strength of ships framed on the Isherwood system has not yet been demonstrated. Section moduli and computed stresses in ship work are of value only as a means of comparison. If the premises upon which we base the calculations are not similar, the comparison fails.

In computing the section moduli of ships framed on the transverse system, we are accustomed to neglect intercostal longitudinals. If, however, the longitudinal framing of an Isherwood ship be neglected because it is intercostal between bulkheads, we penalize that system of framing. On the other hand, it seems obvious that proof should be submitted of the efficiency of bracketed connections between longitudinal frames and bulkheads before full value can be allowed for the scantlings of longitudinal frames. In this connection it should be noted that a very slight yielding of the bracket connection (about .0025 inch) would result in a transmission of stress from the frame to the shell, and it is especially to be noted that this transference of stress would occur long before the metal reached its elastic limit. In fact, unless it can be proven that no infinitesimal slipping occurs in bracket riveting, and, in addition, that the modulus of elasticity of the longitudinal through the connection is at least equal to the modulus of the uninterrupted metal, the assumption of similar premises fails.

This statement of facts seems perfectly obvious, yet, in so far as I know, adherents of the Isherwood system of framing have not publicly submitted data bearing on this important feature of that system of design. In laying out shell and deck plating, we are accustomed to break joints, but under the Isherwood system all longitudinal frames are interrupted at the same girth line, and that, too, coincident with the line of closely spaced rivet holes incident to bulkhead installation. Apparently here is a line of weakness where compensation must be provided, and, as heretofore stated, the sufficiency of such compensation should be demonstrated.

An an indication of the relative importance of the points here raised, the following is submitted. The section moduli of the ships under discussion have been given as

	Hogging condition.	*Sagging condition.*
Transversely framed vessel	161,900	178,100
Isherwood vessel	170,300	189,700

Information was not given as to the exact material assumed in strength. Independent calculations, however, give results in very close agreement with the foregoing if the longitudinal frames of the Isherwood ship are included in strength and if for both ships the decks all athwartship are considered effective, and if one-seventh of the tension material is deducted for rivet holes. For the transversely framed ship intercostal longitudinals are ignored.

Computations have also been made deducing the section moduli for the Isherwood ship, allowing 90, 75, 50 and 0 per cent effectiveness for the longitudinal framing, on the tension side, the above percentages being applied after the one-seventh deduction for rivet holes.

Following this and acting upon the assumption that the longitudinal frames might properly be given 100 per cent effectiveness in compression, section moduli were computed, allowing full weight for compression frames and 90, 75, 50 and 0 per cent effectiveness for the longitudinal frames in tension, with rivet-hole deduction as before.

In all of the foregoing computations, deck plating and longitudinals underneath the deck were included all athwartship. The section computed is located just abaft the break of the bridge, and inspection of the deck plans reveal that only a very short space intervenes between the section computed and a deck opening, 21 feet in width. Obviously neither the deck plating nor the longitudinals are able to transmit stress through the open hatch, nor does the slight increase of .06 inch in thickness of the passing strake compensate for this loss.

A second set of computations was therefore made, eliminating entirely a 21-foot breadth of deck, both the upper and second decks being affected. The one-seventh deduction from tension material was allowed throughout this series in exactly the same manner as before.

The hatch deduction reduced the section moduli for the transversely framed vessel to 129,400 for the hogging section and 143,600 for sagging. The comparative figures for the Isherwood vessel, full effectiveness being allowed for framing, were 128,500 and 142,400, slightly under those for the transverse ship. As before, percentage allowances were then made for the framing, and a series of section moduli deduced. The results of both series of calculations I have correlated in a table, and have also plotted a set of curves of section moduli on a percentage base, the percentage referring to the proportion of longitudinal frames included in strength. The table referred to will be found on page 79 and the curves on Plate 38.

There are four section moduli plotted for the transversely framed ship. These appear as straight lines parallel to the base. In order, they are No. I, the sagging modulus for section with no deductions made for deck openings; No. II, the hogging modulus with no deductions; No. III, the sagging modulus computed with hatches deducted; and No. IV, the hogging modulus with hatches deducted.

For the Isherwood ship there are eight curves. These, too, appear to be practically straight lines but with varying slopes. Curve V is for the sagging modulus, no deductions being made for hatch openings and percentage deductions being made for framing in tension only. This curve lies wholly above curve I, indicating that if the assumptions upon which it is based were valid, the Isherwood ship would be better able to resist sagging than the transversely framed vessel, regardless of the efficiency of frame brackets of members in tension. Curve VI is for the Isherwood ship sagging, the percentage allowance being applied to all frames whether in tension or compression. It crosses curve I at about 71½ per cent, indicating that if hatch openings are suitably compensated for and if all longitudinal framing is better than 71½ per cent efficient, then the Isherwood ship would be, under these circumstances, the better able to resist sagging.

Curves VII and VIII are for the hogging moduli under similar assumptions. They cross curve II, the corresponding curve for the transverse ship, at 65 per cent and 71½ per cent, respectively.

Coming now to curves IX to XII, for which the hatch openings have been deducted, they are found to lie wholly below the corresponding curves III and IV for the transversely framed ship. This would indicate the Isherwood ship as built to be inferior in longitudinal strength to her sister framed on the transverse system, provided, however, that if the longitudinal framing is actually 100 per cent efficient in both tension and compression, the moduli resulting approach so nearly to those of the transverse ship that corrections of slight inaccuracies in the curves might easily make the longitudinally framed ship equal or slightly stronger than the other.

In any event, the curves cast doubt upon the claims of the Isherwood ship to superior longitudinal strength, and if it were found necessary to add material to make it equally strong, the margin in weight saving would be lessened and might be caused to disappear.

Mr. J. W. Stewart, *Member* (Communicated):—I have read with interest the paper given by Mr. John Flodin, namely, "Analysis of the Isherwood System of Ship Construction."

This paper is of interest to me, particularly so as Mr. Flodin has developed it entirely from the merits of the case and has not been in collusion with me on his most clear and interesting treatise of the various features of comparison between the two systems.

TABLE OF SECTION MODULI—DECKS ALL ATHWARTSHIP INCLUDED IN STRENGTH.

	Transversely framed ship.	Isherwood ship.		Percentage of long. fr's figured in strength.
		Percentage applied to all longitudinal frames	Percentage applied to fr's in tension only. Compression fr's are at 100%.	
Hogging	161400 *	170700 ‡	170700	100
Sagging	178900 †	189000 §	189000	
Hogging	......	167500	168300	90
Sagging	......	185500	188000	
Hogging	......	163300	164500	75
Sagging	......	180200	186900	
Hogging	......	155200	158300	50
Sagging	......	171700	184800	
Hogging	......	139600	145600	0
Sagging	......	154300	179900	
Openings in decks allowed for.				
Hogging	129400	128500	128500	100
Sagging	143600	142400	142400	
Hogging	......	126200	126700	90
Sagging	......	139700	141700	
Hogging	......	122700	123800	75
Sagging	......	135800	141000	
Hogging	......	116800	119100	50
Sagging	......	129300	139400	
Hogging	......	105000	109600	0
Sagging	......	116200	136100	

The corresponding moduli as given in Mr. Flodin's paper are *161400, †178100, ‡170300, §189700.

NOTE: The section moduli appearing in this table were computed from measurements lifted from the lithographed midship sections.

Referring to the comparison of weights and capacities, this saving as calculated by the writer of this paper, namely, 142 tons, would appear to be about right. But I observe that he makes no allowance for rivet heads or liners.

With regard to these two items the difference in rivet heads on the two systems would not amount to anything worth speaking of. But the liners, undoubtedly, are less in the Isherwood system than in the transversely framed vessel, and I should say to the extent, in the case he has under consideration, of about 1 per cent. This will be appreciated by referring to the midship section and considering that the framing is 2 feet 2 inches apart, whereas in the Isherwood system it is 12 feet apart. The extent of the saving to this end would probably be close to 20 tons.

There is another item of saving which has escaped Mr. Flodin's notice. This will also be appreciated by referring to the section—that is, the amount of cement chocks that are required between the frames in the transversely framed vessel, whereas in the Isherwood system the necessity of this cementing does not exist.

The saving in this weight would depend upon the amount of coke breeze, or like material, that may be fitted in with the cement, but I should say it must be worth about 10 tons.

These savings, added to the amount as calculated, namely, 142 tons, would increase the deadweight carrying capacity of the vessel and also the saving in weight. The percentage of saving would then be relatively higher than indicated in the paper.

With regard to his remarks where he has touched upon the tankers, it may be advisable to point out that the saving in weight in these vessels run up as high as 15 per cent and even more.

This varies with the bulkheading arrangements. This is now appreciated sufficiently that over 90 per cent of the tankers building are on the Isherwood system and that saving in weight is obtained without impairing the strength of the vessel. In fact, the section modulus of the Isherwood system is almost 20 per cent better than the transverse system, and to that extent the vessel's longitudinal strength is increased.

Regarding the failure of bottom through collapsing, the figures Mr. Flodin has taken out are on the basis of floors on every frame in the case of the transversely framed vessel. As it is evident and well known that there is considerable surplus of strength where floors are fitted on every frame, classification societies have agreed that floors can be fitted on every third frame with bulb angle or channel intermediates. Further, the frame spacing has been approved up to 36 inches, and Lloyd's has allowed a concession of the thickness of the bottom plating on this construction as well as in the case of the floors on every frame.

Regarding the longitudinal strength or bending strength, which Mr. Flodin has so carefully and diagrammatically set forth, it appears that 5 per cent advantage to the Isherwood system was considerably on the low side. In fact, from calculations I have made, the section modulus shows about double that amount.

There is one thing that probably he has given the transversely framed vessel the advantage of, and that he has doubtless left out of the Isherwood vessel. I refer to the continuous deck girders. In a strict comparison, of course, the deck girders should be treated alike in both cases, that is, either included in the Isherwood vessel or excluded in the transversely framed vessel.

Regarding the strength of the decks it would be necessary to have the figures before me so as to exactly see how the comparison was made for deck loads. I am not, however, in a position to discuss this point in detail, but I am certain that the decks of the Isherwood vessel are as well supported to sustain a similar load as the transversely framed vessel. This has been borne out by experience.

Regarding the shell plating, it is not correct to say this is in accordance with Lloyd's rules. The material in an Isherwood vessel is better distributed and disposed in the right direction. This is conceded by classification societies, and they have found it possible to reduce the plating in Isherwood vessels slightly.

Regarding the other matters of strength, I have not had sufficient time to digest all this, but the writer has shown that he intended to find all the weak spots, has studied the pros and cons very carefully, and has "weighed them in the balances" but has found the Isherwood vessels *not* wanting.

Regarding the practical considerations, it is interesting to notice that several shipowners have given Mr. Flodin some very interesting views on the Isherwood system. I am particularly interested in the reply of W. R. Grace & Company with reference to the repair damage to the shell or bottom—that they have allowed the repairers to cut the longitudinal to suit the repair. This is as it should be dealt with, for there is no reason why the strength of the vessel should be in any way impaired if care is taken in fitting an adequate butt-strap where the longitudinals may be cut. I mention this because I have from time to time found that surveyors have endeavored to point out that the Isherwood system is bad to repair because the longitudinals have to be cut back to bulkheads and original butts. There is no reason, of course, for this, and I am pleased to learn one firm at least appreciates this point. I am convinced that there is not an easier vessel to repair than the Isherwood type.

One point on which the writer might have obtained information with advantage is the question of fabrication. There is no doubt, to quote the words of Mr. Piez, that "the Isherwood system lends itself more readily to fabricating than does the vessel of the fish-bone type."

I was pleased to observe in the paragraph before the conclusion that the writer had raised a point which I am quite sure is not generally considered and is deserving of attention. No doubt this is quite a point in the construction of an Isherwood vessel and an important factor in reducing the working cost of a yard.

But there is another point that he might also have called attention to, which reduces working expenses, that is, less furnaces are required, as practically no frame bending is necessary in the Isherwood type of vessel.

THE PRESIDENT:—Are there any further comments on the paper?

MR. JOHN REID, *Member:*—Gentlemen, I hold no brief for Mr. Isherwood, and I am not sure that he requires anyone to sound the praises of his activities in shipbuilding, and certainly it is not my purpose to attack his system—I think it has made good. Mr. Hiatt, in his remarks, rather made out that it had not, but everybody knows that is nonsense. The Isherwood system has made good, and that even in destroyers, and made good under the most difficult conditions, but like every other innovation, it started out with too much of a blare of trumpets. You must put all you have behind a new project, to get through the "stick-in-the-mudness" of the average naval architect, trying to save his owners from making a mistake. Isherwood had to drive through all that; he had to drive through it by ignoring the naval architect and going to the owner and showing him why.

Isherwood in his plan saved steel, he got strength in nearly every direction where he wanted it, and anyone who saves steel today, when you are building 8,000,000 tons of ships in a hurry, is a public benefactor. So he was able to get shipyards, not too highly experienced in shipbuilding, working at the problem of shipbuilding, and almost in twenty-four hours, so to speak, had got out all the scantlings which were necessary, so as to order steel and go ahead with the work—you know what that means.

The Isherwood system can be used in a great many ships today, but it cannot be universally used to advantage. I should not use that system myself on the Lakes, for instance. The reason has to do with repairs. If you run Lake steamers in the Canal, or the big 600-footers, carrying 15,000 tons or more—if you run them in shallow water and constantly rub the bottoms, you will give the Isherwood system very extensive damage, not in any one place, but running quite a distance along the bottom, and this is true particularly in the canals. You cannot use them in the canals, because in the Canadian canals, for which I have built a number of vessels, the bilges are constantly touching the banks and the sides of the ships are in contact with the locks because the locks restrict the dimensions of the boat, and the shipbuilder in trying to improve the dimensions has made the boat narrower; you will have long scores between the longitudinals and constant trouble. I would not recommend the Isherwood system for that kind of work. Mr. Isherwood may have different views, but that is the idea I have formed from my experience.

I should not recommend this construction in connection with very large passenger steamers, because you have a large number of decks which have to go in anyway, and therefore you have ample longitudinal strength in the association of these decks with longitudinal bulkhead, all forward and aft, and there is no reason why you should "Isherwood" these steamers.

I saw a reference in this paper to the Storstad. The Storstad, as you know, sank the Empress of Ireland in the St. Lawrence River some years ago with a terrible loss of life. I had occasion to investigate the collision for the owners of the Storstad, particularly with a view to finding out what happened, judging by the damage done to the Storstad. The Empress of Ireland was under water, and we checked up a good deal of what happened by examining the damage to the Storstad. The one thing I want to bring to your notice is this—the Storstad had rammed the Empress of Ireland a heavy blow at an angle around 40 degrees, glancing past the Empress of Ireland and cutting a broad hole in the Empress of Ireland down through all the decks. The Storstad had only damaged about 12 feet of her own length out of a total length of 425 feet; in other words, the longitudinal framing stood out against this ramming strain and resisted it in a way that no transverse framing would have done, the Storstad not being damaged in the collision bulkhead and not up to it.

So, as regards damage in collision, the one boat gets off well—if it is the rammer. As to the other fellow of course, well, he ought to get out of the way, I suppose.

That brings up the question of repairs. There is very little repairing to be done in a case of that kind. The bow of the Storstad was cut off by running a torch around the hull, the piece dropped off, and a new bow put in. I believe that in a transverse ship there would have been a concertina effect and that the damage would have extended back to the bulkhead, and that very much more serious damage would have occurred.

Mr. E. A. Sperry, *Member*:—I want to call attention to one caution that, it seems to me, ought to be exercised with reference to the construction of longitudinal framed ships, and that is with reference to the torsional factor. The foresquare modulus must receive its principal backbone from decking. Now in all modern cargo ships, where the hatches have to be such a large proportion of the total decking, you have to go carefully with regard to any weakening of the ship against excessive torsion. This was called to my attention when we had the stabilizer on the Worden. There was a very unusual condition. The gyro was used on one occasion to roll the boat and rolled it tremendously in perfectly still water. We thought we noticed quite a difference in rolling on the two ends of the ship, and so we erected battens and with a dock nearby—not too close to interfere with the wave action—very accurate measurements were taken. It was amazing what a difference in torsion would occur between the two ends of that structure, and so I started to investigate. It was a rather small destroyer. We had it loaded to about a thousand tons at one time. I found a very strange thing about torsional strength generally. Now we all know the enormous torsional strength of a tube. Suppose we had a 6-inch tube, 1/8 inch thickness, and, we will say, 20 feet long. You understand that a torsional effort applied at one end could be taken off at the other end of the tube with a very slight deflection. A very severe torsion would in fact show little or no deflection.

Now what do you suppose occurs if we should cut that tube longitudinally into two equal parts? We simply would have two gutters. How does the torsional stiffness of the gutter compare with the torsional stiffness of the original tube? That interested me. We found that the ratio is about 3,600 to 1—that is all. As you know, an ordinary eaves gutter, laid on the ground, of its own weight will show an angle of 90 degrees or more. Here we have a ship—and a ship is nothing but a gutter—just about half of a tube, and so the constructor must look to it that he provides against abnormal fiber torsion.

There is another point which has interested me considerably since we have constructed two pallographs for the Construction Bureau. The author speaks of the natural period of vibration of the hull. We have never found there is any such period from the use of a pallograph. We find, when you change the period of application of the oscillating disturbance, the ship will respond to almost any period before the nodes in the ship will change very markedly. You can run her nodes from a position near the stern to a position forward of the middle of the ship by simply changing the period, so I have an idea that a long, slender structure, such as the Isherwood ship, will probably respond in some way to almost any period that is applied to it, and in the case of some periods, when the nodal points are near the middle of the ship, it will of course respond more readily than the vibrations of a higher period.

Mr. Hiatt:—I would like to inquire if there are any shipbuilders present who can give any further light on the amount of saving in weight they have found to exist in the Isherwood system.

The President:—Can anyone shed any further light on the subject of weight? No doubt that question will be answered later in the written response which will be sent in on the discussion. Mr. Flodin, have you any further comments to make in rejoinder?

Mr. Flodin:—In regard to Mr. Hiatt's remarks as to the financial success of the ship, the paper was not intended to reflect on the conservation of financiers. I am not much of a gambler myself, but, on the other hand, I think a majority of you will agree with me that there is a little too great conservatism among shipbuilders.

Furthermore, Mr. Hiatt points out the weakening of the longitudinals where they are cut at the bulkheads. Undoubtedly there is a weakening, but also, as I have already mentioned, I allowed the same reduction in strength for the longitudinals as in the case of the shell. Furthermore, the classification societies now allow all longitudinals to be cut at the butts of the shell platings; that is to say, the longitudinals are the same depth as the shell plates, less the width of the butt lap, and are strapped at the shell butts. This construction facilitates erecting, because the longitudinals may be shop-riveted to the shell plates. The butt straps are designed to give the same longitudinal strength as the longitudinals, but even so, there is always danger of slipping at the strap. If the classification agencies are willing to allow this, we have an indication which will warrant the belief that the brackets mentioned do not seriously reduce the strength of the longitudinals.

When I first started this work I realized that the brackets were more or less open to question. If I had had the time and money available, I should have preferred to conduct a series of tests on them. I do not know of a machine built by means of which we could imitate the longitudinal stresses induced in a ship, but it would have been well if some sort of tests could have been made to show just what the effect of the brackets is. However, lacking these tests, the best I could do was to give the analysis in its present form.

As to the weakening at the hatch openings, Mr. Hiatt fails to mention that the transverse coaming at the hatch end is stiffened by 10-inch by 3½-inch by 74-inch channel face bar for the second deck and a 10-inch by 3½-inch by 56-inch channel face bar for the upper deck, a face plate being fitted in each case. The deck longitudinals which stop the hatch are bracketed to this transverse, and consequently the transverse may be regarded as a simple beam bent horizontally. Some of the stresses in the longitudinals will be transmitted to this transverse beam and from there to the fore-and-aft coaming, and to the longitudinals at the other end of the hatch. What percentage of the stresses are thus transmitted I do not know, but I do not think it is fair to throw out the entire amount of steel in the longitudinal members between the hatches. I should like to emphasize further that the calculations are meant to be comparative only, so that no injustice was done since the steel between the hatches was included for both ships. Unfortunately our knowledge of stresses in ships or in other structures is not thorough enough to enable us to draw conclusions with any real assurance as to the exact stresses that will be present, and how they will be distributed. If we had such knowledge, we could probably lighten our ships materially and say to the financier:—"Here is a ship. It is absolutely safe, and I can prove it to you here and now, and no need of tests." Let us hope that some day we shall arrive at that stage of development.

As to Mr. Sperry's and Mr. Reid's remarks, I wish to thank these gentlemen very much. They were very illuminating.

Mr. F. M. Hiatt (Communicated):—The remarks offered by the writer in the discussion of this paper were apparently received by some as a criticism of the Isherwood system of framing. The intent, rather, was to question the sufficiency of the evidence so far submitted in proof of superior longitudinal strength.

Attention was invited to the fact that in a measure, at least, the Isherwood system of framing has already passed the test of time. This fact is recognized, but, of course, has no bearing on the relative strength of that type of framing as compared with the transverse. It should be understood that the weaker of two ships may be strong enough, in which case the stronger might properly be allowed a reduction in scantlings.

The transverse system of framing has been built up through years of experiment in seagoing vessels. At present it is the best criterion available as to the requirements which must be met in a seagoing hull. It is therefore eminently fitting that new systems of framing should be analyzed and compared with the transverse system as a standard. It is at the same time of especial importance that such analyses should be fair and just, otherwise self-deception results.

Devices for measuring the working strains in engineering structures have been constructed and are constantly being made more efficient. Comparative results of strain studies for similar ships, with different systems of framing but under similar conditions of loading, would be interesting, and would, no doubt, throw additional light upon this important subject. The writer had hoped that his remarks might elicit information either from some such source or from actual tests of frame bracket connections, and it was with this hope in mind that they were prepared.

THE PRESIDENT:—Mr. Sperry's suggestion affords food for reflection. He is a very chivalrous gentleman, and would be the last man in the world to call a lady names. In the good old days when ordnance experts wished to take a fall out of the naval architect, a ship was called nothing but a "gun platform." Mr. Sperry has gone far beyond that and called a ship nothing but a "gutter." He has added insult to injury, and has also tended to detract from the splendid achievements of the torpedo-boat destroyers in the great war by saying, inferentially, that they had nothing but a tin roof, since he referred to the "gutter" being attached to the "eaves." However, it is refreshing in scientific discussions to have this little by-play occasionally, and I feel quite sure that Mr. Sperry would be the very last to use a wrong characterization of a lady, even though she be a ship.

We have reached the conclusion of our morning program. We will now take an adjournment until 2.15 o'clock this afternoon. The Members of the Council are requested to meet near the rostrum immediately after the adjournment of the morning session.

The meeting then adjourned.

SECOND SESSION.

THURSDAY AFTERNOON, NOVEMBER 13, 1919.

President Capps called the meeting to order at 2.20 o'clock.

THE PRESIDENT:—The Council has recommended the election of an additional number of members, and the list will now be read by the secretary.

SECRETARY COX:—At the Council meet ng at the close of the morning session the following applications for membership in various grades were passed upon favorably:—

Members (6).

Glennard C. Decker, Draughtsman, Hull Division, Cox & Stevens, 15 William St., New York, N. Y. P. O. address: P. O. Box 254, Prince Bay, Staten Island, N. Y.

Richard J. Kingston, Surveyor to the American Bureau of Shipping. P. O. address: 4903 Cedar Avenue, Philadelphia, Pa.

Charles Lang, New York Manager, C. H. Wheeler Mfg. Co., 114 Liberty Street, New York, N. Y. P. O. address: 348 Jefferson Avenue, Brooklyn, N. Y.

Charles Piez, President, Link Belt Co., Director-General, Emergency Fleet Corporation. P. O. address: 910 Michigan Avenue, South, Chicago, Ill.

William E. Quimby, President, William E. Quimby, Inc.; President, Sundh Electric Company; President, Simplex Ref. Company. P. O. address: 209 Parkhurst Street, Newark, N. J.

Alexander Robertson, Chief Engineer, Newburgh Shipyards, Inc., Newburgh, N. Y. P. O. address: 44 LeRoy Place, Newburgh, N. Y.

Associate Members (5).

John J. Eason, Assistant Head, Steel Ship Section, Ship Construction Division, Emergency Fleet Corporation. P. O. address: U. S. Shipping Board, E. F. C., 140 North Broad Street, Philadelphia, Pa.

Joseph Mortimer Kiernan, Lieutenant (J. G.), Construction Corps, U. S. Navy. P. O. address: Hull Division, Navy Yard, New York, N. Y.

Harris Livermore, President, Shawmut Steamship Co., member firm of Wm. H. Randall & Company. P. O. address: 46 Front Street, New York, N. Y.

Charles Randolph Page, Treasurer, Atlantic Gulf and West Indies Steamship Co. P. O. address: 11 Broadway, New York, N. Y.

Clayton Slawter, Draughtsman, Machinery Division, Cox & Stevens, 15 William Street, New York, N. Y. P. O. address: 46 Front Street, New York, N. Y.

Junior Members (2).

Julian Sage Burrows, Assistant to the Superintending Engineer, Pan-American Petroleum & Transport Company, 120 Broadway, New York, N. Y. P. O. address: 136 W. 44th Street, New York, N. Y.

Leonard Ramsey Duncan, Inspector of Repairs, Morse Dry Dock and Repair Company. P. O. address: 6316 4th Avenue, Brooklyn, N. Y.

The President:—Gentlemen, you have heard the list read. What is your pleasure?

Mr. Spencer Miller, *Member of Council*:—I move that the list of applicants as approved by the Council be elected to membership in the various grades.

The motion was duly seconded, put to vote and carried.

The President:—On Saturday evening, at the Waldorf-Astoria, at 7 o'clock, the annual banquet will take place. The secretary suggests this matter be brought especially to the attention of the members of the Society, as space is limited at the present moment. In fact, the overflow is now going into the gallery. Those who wish to make reservations will please communicate with Mr. Kain in the rear of the hall.

The next paper on the program is entitled "Economical Cargo Ships," by Mr. A. J. C. Robertson, Member. Mr. Robertson will present his paper.

Mr. Robertson presented the paper.

ECONOMICAL CARGO SHIPS.

By Alfred J. C. Robertson, Esq., Member.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

The circumstances of the late war and the necessity of largely rebuilding the shipping trade of the world have called the attention of naval architects to the need of a careful mathematical and scientific analysis of the cost of operation of ships, and, as a result, several investigators have recently given us the benefit of valuable papers on the subject. Particular reference might be made to the paper by Mr. John Anderson to the Institution of Naval Architects, London, March 20, 1918; by Alexander Urwin to the Northeast Coast Institution of Engineers and Shipbuilders, January 14, 1919; and by Messrs. G. S. Baker and J. L. Kent to the Institution of Engineers and Shipbuilders in Scotland, February 18, 1919.

Mr. Anderson, in his paper, devotes special attention to working out a theory of operating efficiency based on the number of cargo derricks and winches that can be used and the derrick speed which can be employed in loading or unloading vessels. Mr. Urwin tabulates in a very methodical manner the various elements that go to produce an efficiency figure, and has given us one or two examples worked out. Messrs. Baker and Kent have also treated the matter in this way, but, in addition, give considerable attention to the actual ship form for economical movement of freight.

In the opinion of the writer greater progress could probably be made if naval architects were able to place before ship operators the actual operating costs under certain assumed conditions and with various combinations of size, speed and form of ship. Figures compiled in this way showing the cost of operation for certain German ships on four stated routes were published in *Shipbuilding and Shipping Record* of January 2, 1918, and figures have also been compiled by the United States Shipping Board covering the operation of certain ships over various routes. In each case, however, these figures were for vessels varying in their general characteristics, and it appears to the writer that any true comparison can only be made if the variation in the characteristics of the vessels is directly limited to that feature which it is desired to investigate. He has therefore prepared some calculations in this direction and takes the liberty of placing before the Society of Naval Architects and Marine Engineers such figures and results as he thinks may prove of value.

The method adopted has been to take vessels of four different sizes, of standard proportions, and to work out on the same basis the first cost and the cost of operation for each vessel at each of four different speeds. In addition, the writer has investigated particular features in the cost of operation for one or more sizes of ship. The scope of this series of calculations for an assumed length of route of 3,500 nautical miles between fueling ports is shown herewith, each size of ship being investigated for 10 knots, 11 knots, 12 knots and 13 knots sea speed, using oil as fuel, except as noted:—

A. Block coefficient chosen for each specific speed.

1. With minimum port detention; length of ship 350 feet, 400 feet, 450 feet, and 500 feet.

2. Minimum port detention, coal burning; length of ship 500 feet.

3. Minimum port detention, concrete ship; length of ship 500 feet.

4. Double port detention; length of ship 350 feet and 500 feet.

5. Minimum port detention, with 10 per cent increased resistance, 500-foot ship.

B. Fixed block coefficient for all speeds.

6. Minimum port detention; fine model 500-foot ship.

7. Minimum port detention; full model 500-foot ship.

Each of these ships was investigated for fuel rates of $6.00 and $12.00 with tonnage dues, and with a fuel rate of $6.00 figures were also calculated without tonnage dues. In addition to the above calculations, another series was made with a steaming distance of 7,000 nautical miles between ports, but as the character of the results was the same throughout, the investigation for this length of run was not made so complete. Such figures as the author thinks would be of interest for the 3,500 nautical mile haul are given herewith in Plate 39.

In order to extend these calculations, a very considerable number of assumptions had to be made, as follows:—

(*a*) *Proportions*.—The length of the vessel was taken as the water-line length, which would also be the length between perpendiculars for twin-screw vessels. The length between perpendiculars for single-crew vessels would be somewhat greater than the figures given.

The breadth molded was taken as equal to 0.10 of the length on water line, plus 14 feet, this proportion giving ample beam for stability even with a considerable amount of superstructure.

The load draught in each case was chosen to give a breadth upon draught ratio of 2.25, which is approximately the best ratio for economical propulsion.

(*b*) *Speed and Power*.—The speed in each case is such as could be maintained in any ordinary weather at sea. The power was arrived at in each case from tank test data, a propulsive coefficient of 52 per cent being assumed and a margin of power allowed to maintain the required speed at sea. A discount of 20 per cent from the figures given for indicated horse-power should be made to arrive at the horse-power on a smooth-water trial trip fully loaded.

(*c*) *Weight of Ship*.—The weight of the steel hull has been found to vary slightly with the block coefficient, and this variation has been allowed for. The weight of outfit corresponds to a first class specification and has been taken as identical for each length of vessel irrespective of speed. The engine-room weights are taken as varying directly with the horse-power, taking 4 horse-power to each ton of engine-room machinery. The total deadweight was assumed to be the difference between displacement and the sum of steel hull, outfit, and engine-room weights.

(*d*) *Cost of Ships.*—The cost of the steel hull was assumed to vary directly with the weight of steel used, and the cost of outfit was taken as constant for each size of ship. The cost of propelling machinery is based directly upon the horse-power developed; and the basis of these costs was so chosen that the 400-foot ship, at 11 knots speed, would show a cost of approximately $150.00 per deadweight ton.

(*e*) *Weight of Fuel and Stores.*—The fuel oil consumed was figured at one pound per indicated horse-power per hour, plus a fixed amount for each size of vessel in port. The port fuel consumption allowed for minimum detention was taken as 20 tons for the 350-foot ship, 30 tons for the 400-foot ship, 40 tons for the 450-foot ship and 50 tons for the 500-foot ship. A figure was taken for fresh water, make-up feed, and consumable stores equal to one-half the weight of the fuel oil required per voyage.

In fixing a fuel rate per ton of oil fuel it is intended that this figure should cover also the cost of lubricating oil, fresh water and similar charges based on the horse-power of the engine. For this reason $6.00 and $12.00 as fuel rates would represent a cost somewhat lower than this for the oil alone.

(*f*) *Port Detention.*—The time in port is of great importance in the estimated cost of operation of a ship. This is a variable figure, and calculations are given showing the effect with minimum detention and with a detention equal to twice the minimum. I have taken a figure for the minimum length of time in which a vessel should be detained in port as depending on the quantity of cargo which must pass through every square foot of hatch. This quantity, of course, with similarly designed ships depends directly upon the depth of the ship, and I have therefore allowed one day in port for every 5 feet in depth of ship, as a minimum port detention figure. In addition to the above port detention, thirty days per annum have been allowed for overhaul.

(*g*) *Operating Expenses.*—The cost of crew is estimated from the United States Shipping Board's rate current at the beginning of this year and includes an allowance of $300.00 per annum per man for sustenance. In addition, a management allowance has been made of $500.00 per month. The cost of manning, however, must be considered as only approximate, as the exact number of crew would depend to a certain extent on design of ship and route.

Depreciation, insurance and up-keep have been figured at 13 per cent per annum on the first cost of the vessel. Tonnage dues have been assumed at $1.50 per net ton register, and the assumption has been made, for the sake of comparison, that the net tonnage in each ship would be equal to one-third of the load displacement in tons.

After figuring out the cost of the ship, its deadweight carrying capacity in tons of 2,240 pounds, and cost of operation per annum, these figures have been reduced in each case to the basis of $1,000,000 investment; and in order to make the figures more easily grasped I have assumed a gross freight rate of $6.00 per deadweight ton for the 3,500 nautical mile run. From this rate, however, I have deducted $1.00 per ton for handling and 15 cents per ton brokerage, showing a net revenue of $4.85 per deadweight ton on this run. For the 7,000-mile run I have assumed the freight rate of $10.00 with a deduction of $1.00 per ton for handling and 25 cents per ton for brokerage, giving a net rate of $8.75.

The profits in operation calculated on the above assumptions have in each case been figured upon the assumption that the ships are operated on each leg of their route with a full cargo. This assumption is only a hypothetical one. The freight rate should therefore be so adjusted that it shall bear the proportion to the assumed rate that a full cargo does to the average percentage of cargo carried per trip. For example, if the ship operated with an 80 per cent load in one direction and a 20 per cent load on the return voyage, 100 per cent load is divided between the two voyages, so that only 50 per cent freight movement is achieved and the freight rate would therefore be doubled to realize the profits shown in the table, Plate 39.

RESULTS

The results of the calculations are shown in a series of diagrams on Plates 40 and 41 and may be readily summarized as follows:—

(*a*) *The Cost Per Deadweight Ton.*—Fig. 1, Plate 40, shows that the cost is sensitive to speed, increasing rapidly in all sizes of ship with increase of speed. Fig. 11, Plate 41, shows that the cost is only slightly sensitive to increase in size, showing tendency to fall with increase up to about 450 feet at 10 knots and a larger size for faster vessels.

(*b*) *Deadweight Moved Per Million Dollar Investment Per Annum.*—Fig. 6, Plate 40, shows clearly that for large freight movement slow speed is essential. At 10 knots a 350-foot ship will move more cargo than any larger vessel. At 11 knots the 400-foot ship is the best; at 12 knots the 450-foot ship is better, and at 13 knots the 500-foot ship will move as much as any smaller size.

(*c*) *Profits Earned Per Million Dollar Investment.*—Figs, 2, 3 and 4, Plate 40, show the profits earned for a 3,500-knot run as plotted upon speed of ship, and Figs. 12, 13 and 14, Plate 41, are corresponding diagrams for the 7,000-knot run. Figs. 7, 8 and 9, Plate 40, show results for the 3,500-knot run plotted on the size of ship.

All these diagrams show clearly that the slow-speed ship is the money-earning ship, and there is a very steady decline as the speed increases, the 350-foot ship earning twice as much at 10 knots as at 13 knots.

It will be noted from Figs. 7, 8 and 9, Plate 40, that at a speed of 10 knots a ship 400 feet long gives practically the very best earning proposition that can be designed, and that for 11 knots a length of 450 feet is probably the maximum economical length. These figures, of course, assume the ship to have a capacity cargo, and even a small reduction in this cargo would tell heavily against the larger ships.

These curves completely justify the building of tramp ships between 350 feet and 400 feet in length, with a sea speed of 10 knots.

(*d*) *Form of Ship.*—To compare the effect of a single design run at various speeds, with designs worked up for the individual speeds, calculations were made which are plotted on Fig. 5, Plate 40. The figures for these calculations are shown in columns 32 to 39, Plate 39.

They show that a vessel designed for a speed of 13 knots, and powered for that speed, will earn more for her owner running up to full speed than at a lower running rate. On the other hand, a vessel designed for 10 knots, with power good for a speed of 13 knots, shows a falling off in earning at speeds over about 11 knots. This diagram clearly shows the advantage of having a ship carefully designed to work under specific conditions.

(*e*) *Value of Model Experiments.*—Fig. 20, Plate 41, shows the actual loss in profit for the 500-foot ship where the power to drive it is 10 per cent above the minimum. This 10 per cent seems large, but I assure you it is a very modest estimate of the increase of power necessary in a model designed without the use of experimental data. Even careful students of ship resistance might produce a ship design of this kind. This year, for example, we were able to lower the resistance of a 400-foot ship at 10 knots by 8 per cent through increasing the displacement by 1½ per cent; and in another case, with two ships of identical size and fullness, a slight change in the curve of areas of afterbody made 6½ per cent difference in resistance without change of displacement.

The cost to the shipowner of this extra 10 per cent is one of the astonishing features of the figures compiled.

At 10 knots the extra power for the 500-foot ship is 325 horse-power, and this gives an actual loss in earning capacity of $16,636.00 per annum, which, at 5 per cent compound interest, represents in twenty-five years 79 per cent of the cost of the ship, or, in this particular case, $1,610,250.00. At 13 knots the cost is even greater; the 583 additional horse-power costing the ship operator a loss in profit of $24,789.00 per annum, equal to $2,782,230 for the 500-foot 13-knot ship in a life of twenty-five years.

(*f*) *Detention in Port.*—Fig. 10, Plate 40, shows for the 500-foot and 350-foot ships the difference in realized profits if these ships are detained twice as long in port as the ship with which they are compared. Every shipowner is aware that the earning capacity of the ship is measured by its time at sea, but these figures will help to emphasize the necessity for greater care being given to the improvement of the methods of loading and unloading ships.

In Fig. 19, Plate 41, the time at sea is expressed as a percentage of the year, and it is shown that the slow-speed ship can remain at sea for a higher percentage of its life, and to this largely is accountable the superior earning capacity of this type of ship.

(*g*) *Application to Design.*—Figs. 16, 17 and 18, Plate 41, give, respectively, size of ship, power of ship, and capacity of ship for the various speeds and sizes dealt with in this paper. These three diagrams should be of some use for preliminary design work to rapidly approximating the dimensions of a new ship. They are only of definite value, of course, if the proportions and characteristics of the ships are similar to those dealt with in this paper.

(*h*) *Fuel.*—Figures compiled for the 500-foot ship indicate that for the same profits the cost of coal fuel could only be 53 per cent of the cost of oil fuel. This figure, of course, includes putting coal in bunkers, and it takes full consideration of the saving in weight and increase in carrying capacity of the ship due to the use of oil fuel. This economy in oil fuel will not be maintained, of course, if the ships are not running with capacity cargoes.

(*i*) *Concrete Ships.*—The 500-foot ship was calculated also as a concrete ship, and the assumption was made, for the sake of comparison, that depreciation, insurance and up-keep would be at the same rate per annum as the steel ship, viz., 13 per cent. On this basis, in order that the concrete ship should show the same operating profits with the steel ship (the cost of machinery being the same), the cost of hull and outfit for the concrete ship at 10 knots should not exceed 79 per cent of the cost of the hull and outfit of the steel ship, and at 13 knots it should be limited to 70 per cent of the cost of the steel hull and outfit. The cost of a concrete ship per ton of deadweight to show equal profits with the 500-foot steel ship, is shown on Fig. 1, Plate 40.

These figures are on the basis of $6.00 fuel and no dues, but they are approximately true for the other working assumptions.

DISCUSSION.

The President:—Gentlemen, Mr. Robertson's paper, "Economical Cargo Ships," is before you for discussion.

Professor H. C. Sadler, *Member of Council:*—Mr. President and gentlemen, I think Mr. Robertson is to be congratulated in giving us a paper of this type. It is unfortunate, in some of our papers, that the writers occasionally omit certain data on which they have based their conclusions. Mr. Robertson has been very careful to give us all the information on which the results of his calculations are based, so that it is a comparatively easy matter, with different assumptions, to reproduce a curve for any given set of conditions as to freight rates, first costs, running expenses, etc.

I may say that the paper perhaps justifies, to a certain extent, the work of the Emergency Fleet Corporation. As you know, a large proportion of the vessels built were of the cargo type, vessels from 8,800 tons up were in the majority, and they had a speed of about 10 knots, showing that they were really of the good, economical cargo type.

There is also an interesting point brought out in the paper, and that was in connection with the size of the ships. We are all apt to be led away somewhat by the thought of the large vessel being the most economical. Fig. 6, Plate 40, particularly, to which Mr. Robertson has already drawn attention, is, I think, most illuminating. It may be a common error to imagine that the larger vessel will move more tons per annum, whereas that is a factor depending entirely on speed, and it is interesting to note that a 350 or 400-foot ship will actually move more cargo than the 500-foot ship at 10 knots. That is a very significant fact.

Another point to which Mr. Robertson has referred, and on which I should like to say a few words, is in reference to the value of the model experiments. I hope I shall not be accused of special pleading. I think most of you know me well enough not to accuse me of that, but I will give you a general idea of what so often happens. A shipbuilding company spends one or two weeks in getting up a set of lines for a ship. The rest of the plans, in the meantime, are going along, and then some time afterwards they feel they would like to have a model tested to find out what power will be required to drive the ship at a certain speed. Of course, that can be obtained from the model results very accurately, but it seems to me that is the wrong way to go about it. I personally have tested a number of models and have been able to show, as Mr. Robertson says here, that I can save 7 to 10 per cent on the same displacement, for the same speed, and when I draw the attention of the shipbuilder to this fact the answer is:—"Unfortunately, the work is so far ahead we cannot change it."

Now it seems to me, if a little more time were given to model testing in the very earliest stages of design, a great deal of economy could be obtained. We can supply the results of what horse-power the ship will take to drive it 10 or 12 knots, but that is not all that is really wanted—you want to have the best ship for the speed, and it is unfortunate, sometimes, that the work has gone so far ahead that changes cannot be made.

The final reference to concrete ships is one, perhaps, that will help supplement Mr. Wig's paper this morning, and under given conditions it is rather interesting to note that there still seems to be hope for the concrete ship, provided the first cost can be kept down.

Mr. E. H. Rigg, *Member*:—This subject is one which has been given more attention during the last few years before the technical societies than ever before. At first sight this may seem curious when we reflect that they were war years, 1919 being still under the influence of war conditions; on second thought this study will be found to be closely allied to the war. Never before in our time has attention been so keenly directed to producing ships quickly. Large programs of standard vessels have been adopted here and in Britain; the opportunity came and a great deal of study of ship economics has been made. While economy as ordinarily understood has necessarily been a secondary consideration in war-time production, it still remains true that these large programs have furnished the opportunity, and indeed have made necessary, detailed studies of ship economics on a large scale. Mr. Robertson's contribution to the literature of this subject is most timely. The 1918 and 1919 published papers alone form quite a comprehensive literature and the Society is to be congratulated on a useful paper which shows on its face a great deal of hard work. It should commend itself particularly to shipowners and operators. How many times is an inquiry for a ship based on some such analysis as given by Robertson in this paper or by Anderson, Urwin, Baker, Kent and Donald in similar papers and read before sister societies in Britain?

Other investigations covering the economics of still larger vessels are available, notably by Sir John Biles and Lord Pirrie, also by the Canadian Royal Commission on Overseas Communications.

Referring to line No. 36, Plate 39, it will be noticed that the cost of manning and management is the same for each speed of any given length of ship. With horse-power for 13 knots practically double that for 10 knots, it is difficult to see why engine-room manning costs were not increased with speed. For double power about a 35 to 40 per cent increase in engine-room salaries and wages would appear to be in order.

The idea of first stating the figures for the actual ship and then reducing them to a basis of each $1,000,000 invested is one that will appeal to the operator and investor at once; ships and shipping are before the public eye as never before, and this method of presenting results is most fortunate.

Fig. 3, Plate 40, gives us a diagram of dwindling profits as speed goes up. It will be of interest to take this another way; that is, increase the freight rate to maintain equal profits as at 10 knots. These figures refer to the 3,500-mile voyage and minimum port detention, oil fuel at $6.00 and tonnage dues at $1.50.

TABLE OF GROSS FREIGHT RATES.

Length of vessel, feet.	Speed of ship in knots.			
	10	11	12	13
350	$6.00	$6.33	$6.87	$7.82
400	6.00	6.27	6.71	7.32
450	6.00	6.24	6.57	7.08
500	6.00	6.25	6.54	6.98

This table shows clearly the advantage of the large ship at the higher speeds; the 500-foot ship can take freight at 84 cents per ton cheaper than the 350-foot ship at 13 knots. Still another interesting comparison is to take a fixed gross annual profit of $200,000 per million invested for all ships and see how the freight rates compare. This table works out as follows:

TABLE OF GROSS FREIGHT RATES.

Length of vessel, feet.	Speed of ship in knots.			
	10	11	12	13
350	$5.91	$6.24	$6.76	$7.70
400	5.71	5.97	6.38	6.95
450	5.67	5.89	6.20	6.68
500	5.61	5.84	6.11	6.51

This table shows that the 10-knot vessel can always underbid the faster, other things being equal; it also shows that the 450-foot ship at 11 knots can underbid the 350-foot ship at 10 knots. This is another confirmation that the tendency to run at 11 to 11½ knots in the larger freight vessels is sound. The penalty paid for high speed in small ships is again brought out.

Mr. Robertson does well to direct attention to the painful results of undue detention in port, and he confirms the conclusions previously found. This again emphasizes adequate handling gear and terminal facilities as a necessary accompaniment to successful operation.

If the deficiency in handling is attributable to the ship, it is the owner's loss; if to the port, while the ship may be able to charge higher freight rates, the port suffers to that extent in the shape of higher prices for the article shipped.

It would be interesting to take the time necessary to compare the different standard ships built in large numbers and analyze them by the methods adopted in this paper. This would be somewhat laborious; but it can be safely concluded that the main points here brought out are confirmed in practice.

Values are changing so fast these days that any set of figures assumed would be incorrect and perhaps radically so in a few months' time. The value of this paper is not affected by such variations to any material extent; it furnishes us with one method of analysis which can readily be corrected for changes in values. The fundamentals of size, speed, power, capacity, etc., remain constant, leaving the variables only to be re-figured from time to time as occasion requires.

Mr. Spencer Miller, *Member of Council:*—I wish to ask the author to point out how one can use his tables to arrive at the cost per ton of waste material carried on any ship; for example, suppose by improved apparatus the water, the coal, or the oil fuel carried on board ship could be reduced by one ton. What is the value of such a ton of weight saved? Suppose that certain machines be made of steel instead of cast-iron and thus reduced in weight to the extent of one ton, how much saving in money will be effected by such a reduction in weight over a period of say ten years?

The President:—Is there any further discussion? If not, we will ask Mr. Robertson to close the discussion on his paper.

Mr. Robertson:—I do not think I have much to say now, except to thank the gentlemen cordially for their addition to the contribution involved in this paper. The information that Mr. Rigg has presented to us has considerably enlarged the interest of the paper.

In answer to Mr. Miller I will say that the figures presented in the paper are drawn up in such a way as to allow an accurate estimate of the effect of a single ton of difference in the deadweight, or in the weight of structure of the ship. It is a mere matter of arithmetic to arrive at what the amount would be.

The following is an example of calculation of cost of one ton in the economical operation of a ship:—

Take Column 4, Plate 39; the cost of operation of this ship is (line 49), per million dollars invested, \$380,440; and the earnings are figured (line 53) at \$175,612, making \$556,052 gross earnings per million dollars invested. The total cost of this ship, however, is only \$784,330, so that the earnings would amount to \$436,128 with a cargo deadweight of 5,299 tons.

Each ton of deadweight therefore will be required to earn \$82.30 per annum, and if one ton were lost in the ship due to uneconomical employment of material, the sum of \$82.30 per annum would be lost under the conditions assumed.

The value of this sum in twenty-five years, the assumed life of the ship, including

5 per cent interest, is $3,928, and this would represent the total cost to the shipowner of one ton of additional weight in the hull. Of course the assumptions on which this is based would not be made good in practice, but how far they would come short would depend on each individual case.

The President:—As previous speakers have stated, Mr. Robertson's paper represents a vast amount of work, the results being compressed into a comparatively small space. It should be particularly helpful to the ship operators. I think the Society owes a sincere vote of thanks to Mr. Robertson for compiling the data and putting it before us in such an expressive way. The Chair therefore takes the liberty, on behalf of the Society, of thanking him for his contribution.

The next paper is entitled "Non-rolling Passenger Liners—Observations on a Large Stabilized Ship in Service, Including the Plant and Economies Effected by Stabilization," by Mr. Elmer A. Sperry, Member.

Mr. Sperry presented the paper.

NON-ROLLING PASSENGER LINERS—OBSERVATIONS ON A LARGE STABILIZED SHIP IN SERVICE, INCLUDING THE PLANT AND ECONOMIES EFFECTED BY STABILIZATION.

By Elmer A. Sperry, Esq., Member.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

Now that it has been adequately demonstrated in the case of a number of important installations that a ship can be guaranteed against all rolling, a great forward step is possible and a new era opened up for the American passenger-carrying service. The traveling American demands the greatest possible degree of comfort, and shipping interests will not be slow to meet this demand by the latest and most up-to-date equipment cast strictly on American lines.

We will have at no distant date the Service-de-Luxe on the Atlantic and probably also on the Pacific, or, in American travel lore, the "Pullman Service of the Sea." Plans are already in progress for an extensive adoption of the new principle, so that the benefits and economies resulting from a ship guaranteed against roll will be available to the traveling public. This great forward step has been made possible by work going steadily on here in America for the past fifteen years.

The point of large-scale demonstration has now been reached, and full stabilizing moments, requisite for preventing all roll of even the largest ships, have been practically developed. The great principle thus finally established is that full stabilizing moments can now be simply and effectively delivered to a ship that is quiescent and entirely free from motion. The motion of the ship itself having heretofore invariably been relied upon to create the quenching moments, anything like full extinction of roll has of course been impossible.

In all previous attempts to prevent rolling, the equipment has operated on the passive principle, depending on a certain amount of roll for the stabilizing moments, as stated; and the amount by which the roll has been reduced has never been satisfactory, nor have the means themselves been practicable. Another difficulty is that the phase lag is found to be insurmountable, the stabilizing effect—such as it is—arriving "the day after the fair."

The active gyro stabilizer solves the problem and works entirely independently of the motion of the ship. Delivering full counter-moments to a motionless ship, it is thus free to deal directly with the wave slopes themselves, *i. e.*, with the rolling increments of the sea as they are in the process of developing, even before incipient rolling has set in. The active stabilizer thus acts as a simple preventive of rolling, working in harmony with the slow period of the ship, yet dealing alertly with each increment as it arrives, irrespective of magnitude or direction; holding the ship most satisfactorily upon an even keel, apparently without effort; demonstrating how little is really required to keep a ship from rolling, or, rather, from beginning to roll, if only a basically preventive method is employed.

With the complete solution thus in hand, accomplished with a small and simple equipment, it is believed that the older passive types, with their great weight, fractional results, and uncertain operation, will become obsolete.

The equipment that brings about this important result is so simple and unique that a brief résumé of its development, the principles involved, and the performance of the non-rolling ship itself cannot fail to be of interest at this time.

The possibility of producing the non-rolling ship was known to the leading naval architects and investigators in England, notably the elder Froude, prior to 1880. A little later Sir Phillip Watts, one of the great masters of this noble art, became convinced from Froude's work that, inasmuch as it takes so little to hold a ship on an even keel and to prevent it from rolling, this very desirable result should be accomplished. As Chief of Construction of the British Admiralty, he, with a corps of engineers, at once started in to establish this great principle and to put it into execution, by building "anti-rolling tanks," as they were called, into the ship's structure and then filling these elongated tanks up to a certain critical point with sea water. These were expected to take on periodic oscillations from the ship's rolling motion, thus creating moments which would always be in opposition to the rolling and tend to reduce it. These tanks were actually built into two war vessels by the Admiralty, and experiments were undertaken with rather discouraging results, on two counts:—

First, it was found that there were required a large tank and a large amount of water, representing quite an appreciable percentage of the total displacement of the ship—such a large percentage, in fact, as to be prohibitive. The other point was the difficulty in maintaining proper phase relation between the natural period of the ship, the period of the water in the tank, and the period of the waves of the sea. More about this further on. It will suffice to say that when this phase relation is not maintained, these tanks can easily become dangerous, tending to increase the roll rather than to reduce it.

Later the great English engineer, Sir John Thornycroft, undertook to accomplish the same result by a large horizontal pendulum operated hydraulically down in the hold of the ship; and the report of Sir William White—one of the Admiralty's greatest naval architects—on the actual operation of this device in a ship at sea in a storm (while Sir John I. and Sir William were dining together) makes one of the notable chapters in the early history of attempts at stabilization of ships. This stabilizer worked well at times, but it was found to have the same difficulty as the anti-rolling tanks, namely, in order to secure a sufficient reduction in roll to make the equipment commercial and practical, the weight of the equipment was prohibitive.

Later, Herr Frahm of Hamburg re-invented Sir Phillip's tanks, but though he made a contribution to the mathematical treatment of the subject and introduced some refinements in manual control valves and also in shape and contour, the tanks were found to be open to the same objections as those of Sir Phillip's.

Dr. Schlick, who had done such wonderful work in balancing marine engines, then came forward with his passive "gyrostat" or "Kreisel," as he called it. This was installed on a small torpedo boat and tested in the North Sea. The Admiralty sent Sir William White, who, by the way, was a friend and admirer of Dr. Schlick, to report upon this device, and Dr. Schlick's rights were taken over by a leading English firm. The report is interesting. Sir William found this to be another device which depended for its action on the initial roll of the ship—that is, the quenching moments were dependent solely upon the ship's motion—and these would have been quite effective as a roll reducer had it not been for the same unavoidable lag in phase, which seriously impaired their efficiency. In any event, the combined difficulties, some of which are cited, were sufficient to prevent the adoption of this device, or, for large ships, even its serious consideration.

Later the action of anti-rolling tanks came under careful review by our own Taylor. This was before he became Admiral and Chief of the Bureau of Construction of our Navy, and while he still had charge of the big experimental tank at Washington, designed and constructed by him, and of the great collateral equipment that forms a part of the experimental and model department of the Navy. In this connection, Admiral Taylor, with his wonderful grasp of all this intricate phenomena, designed tanks that were unique; and to make his research and investigations complete he went at the matter in the most thorough manner, not confining his observations to one tank alone, but going to sea with a ship equipped with no less than three tanks. This was the most complete equipment, and the investigation was one of the most searching into the action of anti-rolling tanks that have ever been made the world over—as is characteristic of all of Admiral Taylor's work, which is conceived from a most complete and comprehensive standpoint and carried out in the most rigid and searching manner.

It is difficult to believe that the great art of naval architecture appreciates to the full the debt that it owes Admiral Taylor. Some little glimpse is had of the relation that this distinguished authority bears to the world-wide art, when we remember that at the time the Hawk, warship of the British Admiralty, hit the Olympic near Southampton, within four hours of this memorable collision both sides had tried to reach Admiral Taylor by cable to retain him in the suit that they knew was imminent as the result of this accident. So the Government released Admiral Taylor to the Admiralty in order that important precedents might be established in the maritime world.

But to revert to Admiral Taylor's anti-rolling tank experiments. In this connection he reached certain conclusions, as he always does. These conclusions are interesting, as here we have the real crux of this whole matter. His verdict was that outside of the prohibitively large weights and extremely important athwartships space occupied by the tanks, neither the United States Navy, nor in fact any navy or marine, possessed the personnel requisite to keep the period in phase relation with the natural period of the ship and also with the period of the sea. So at last this phase of the art has had the extremely great advantage of passing under critical review by probably the greatest living authority on these matters.

The difficulty with all of these attempts on this most important problem can be reduced to an extremely simple proposition:—With all of these methods of attack, a pound weighs only a pound and its effectiveness is measurable in simple terms of the lever-arm of its application, or the distance from the center of oscillation of the ship at which the moments become effective. This is the reason for the excessive and even prohibitive weights necessarily present in all these methods of reducing the roll of ships.

Now what is wanted is not the reduction of roll, but the actual prevention of all rolling of ships, and it is just here that the powers and resources of the active gyroscope step in. For years engineers have observed the strange peregrinations of the gyroscope, but have failed to perceive the enormous powers that lie dormant in this simple apparatus, only awaiting the application of artificial "precession" to place it under perfect control and to render it abundantly serviceable for stabilizing even the largest ship. By the way, the larger the ship is, the easier it is to stabilize her.

In the gyro we have a most unusual illustration of the phenomena that Cicero classed among "real blessings"; that is, when nature's laws work to aid rather than to obstruct progress, by no means a too frequent occurrence in engineering. The gyro outclasses all other mechanisms in that, while its weight and cost vary as the cube of a lineal dimension, its stabilizing power varies no less than the fifth power. This runs into a very great gain in gyro stabilizers for large ships.

Why is the gyroscope more available for solving this great problem than the tanks on the one hand or the great pendulum of Thornycroft on the other? The answer is simple. Whereas in the prior art, as we have stated, a pound is only a pound, in the gyroscope a new and extremely far-reaching situation is created. In arriving at the powers available to stabilize the ship with the gyro, every pound is multiplied by the velocity of the particle, so that a comparatively few pounds are actually capable of doing the work of tons. With the active gyro this is all held in phase, and, as described, is available for the important purpose of holding the ship free from even the beginnings of roll.

Those unfamiliar with the subject, and even some naval architects, have feared that the forces and stresses involved might endanger the ship's structure in case of a heavy storm. The exact nature and magnitude of these stresses are perfectly well understood and have now been brought under careful observation in quite a large number of equipments in actual service. We are therefore speaking from a wide range of accurate knowledge on this important item, and it will be interesting to know that the conclusions by the highest authorities are that a ship which freely rides the waves with its mast held vertical, being completely stabilized by the little gyro equipment in her hold, is subjected to less than one-fourth and often less than one-sixth of the very large strains present when it is allowed to roll under exactly the same storm and weather conditions and with the heading unchanged in the same sea. With the gyroscopic stabilizer equipment on board, we have the unique situation of being able instantly to throw it on or off, in action or out, at will, by stopping its slow precessional movements, so that we can observe exactly what happens under the two conditions and repeat each condition as often as we choose and hold each under complete observation as long as we choose, under any given sea or weather condition. And just such tests as these have been repeatedly made and studies pursued until they are well known and understood. Thus the presence of the stabilizer on the ship reduces and holds to a very low value the stresses and strains which in the case of an unstabilized ship in storms often rise to high and dangerous magnitudes. Many thousands of feet of record have been made and studied. Fig. 1, Plate 42, shows a photograph of some of these.

Here again Admiral Taylor's work comes to the fore. The development of the active gyro stabilizer was aided materially through the encouragement given by this naval officer. He was among the first to appreciate the important results likely to follow the application of the active principle, *i. e.,* the ability to develop pure torque stresses in ships without change in loading or moving of weights or masses, and to direct these stresses and emplace them at will upon a ship independent of the state of motion of the ship itself and also independent of any particular position of the equipment upon or within the ship. This whole art certainly owes much to Admiral Taylor.

Everyone is familiar with the groanings, creakings, and weird noises that are always present in heavily laboring or rolling ships. These illustrate the stresses and strains to which she is being subjected. Imagine the sensation when the stabilizer is thrown into action and these sounds cease forthwith, positively demonstrating that the heavy stresses have vanished. The stabilizer thus becomes one of the greatest safety devices yet invented, imparting absolute security to the great hull and structure of the ship and materially prolonging its life. All of this, of course, is wholly outside of the consideration of comfort, which is one of the prime reasons for the installation on passenger ships.

In this connection it will be interesting from a technical standpoint to know that some time before the first stabilizer equipment was installed by the United States Navy, the great English naval architect, Sir William White, was brought to this country in consultation on this subject. He stated, after careful review of the facts, that from the naval architect's standpoint the strains introduced by the gyroscopic stabilizer in holding a vessel absolutely free from roll were insignificant, and that if we laid hold of a single frame of an ordinary steel ship we would have a factor of safety of about six, and furthermore that these strains and stresses were only a small fraction of those existing in the hull and general structure of a ship when rolling in a storm.

This great authority went further and stated something that our highest authorities in this country then doubted, namely, that as soon as a ship was stabilized in a storm and rolling was prevented, that ship would not ship seas, but her decks would immediately begin to be dry and would remain dry. Although we did not at the time believe that this could possibly be true, our universal experience since we have had these installations in operation demonstrates the absolute truth of the statement, as the result of many observations and experiments, incidentally showing the great insight of Sir William.

Even some excellent authorities, before actually having the unique and extremely interesting experience of being aboard a stabilized ship, have confused a stabilized ship with a dock, expecting the waves to pound the ship when stabilized, and it is with great surprise and satisfaction that they have repeatedly discovered just the reverse to be true. A stabilized ship invariably rides the sea, gradually rising and falling with the sea with a wonderful degree of gentleness. Her masts quickly come to the vertical, and all pounding and splashing disappear as soon as stabilizing sets in. Fig. 2, Plate 42, shows the original active stabilizer installed on the destroyer U. S. S. Worden.

Other facts have been learned from the performance of the stabilizer in heavy weather. It is found actually to contribute a number of definite economies in the operation of the ship. Anyone who has ever undertaken to pilot a heavily rolling ship and to hold her to her course has realized the enormous amount of "helm" that is constantly required, and the resulting very sinuous course that the ship takes in spite of the best efforts the helmsman can make under these conditions. The diagrams in Fig. 3, Plate 42, graphically illustrate this and other features.

No pains have been spared in studying this important phase of the contribution of the stabilizer. The gyro compass with its enormous directive power enables automatic records to be made of the most minute orientation of the ship. These have been secured and also simultaneous graphic records of the amount of helm being used by the ship, also automatic, so that there could be no question as to exactly what was happening. The study of these records has been full of interest, developing an accurate method of analyzing and aiding to establish the losses under this division.

Fig. 4, Plate 43, shows a characteristic stabilization curse. Fig. 5, Plate 43, shows the two graphic records of yawing, the record to the left being a non-rolling record, and that to the right showing three to four times the yawing due to rolling. Fig. 6, Plate 43, is a simultaneous helm record and corresponds to the left-hand record in Fig. 5. Fig. 7, Plate 43, is the helm record corresponding to the right-hand yawing record. These records were made on a 16,000-ton ship with the same series of helmsmen at the wheel. Fig. 7 and the right-hand yawing record clearly show the three-fold losses due to rolling:—First, the sinuous course: second, the bad angle of attack and the wider path; and third, the direct retardation due to increased helm.

Operating engineers and naval architects know that even a very slight amount of "helm" acts as a tremendous retarder in the forward progress of the ship, and especially is this emphasized when a very large amount of helm has to be constantly employed. This slows down the ship, uselessly wasting a great deal of the propulsive power of her engines. Again, the sinuous course that is invariably steered by a wallowing ship causes it to travel a considerable extra distance, always accompanied with a "bad angle of attack," causing a large extra power consumption. A stabilized ship is practically self-steering. This comes as a sort of a bi-product of stabilization, the stabilized ship requiring practically no helm, regardless of weather.

But there is a still greater source of power waste in rolling ships. As the hull constantly oscillates back and forth, its form-lines encounter and constantly displace laterally, with extra friction of impact, hundreds and even thousands of tons of water, and this persists, going forward with every roll. This, in connection with the extra wetted surface involved, added to the extra stream-line losses and skin friction inpingement, especially when bilge keels are present, amounts to losses of very great magnitude in terms of actual horse-power wasted. Model experiments and resulting calculations indicate that these losses are much higher than have been supposed. Fig. 8, Plate 44, graphically expresses by the shaded area the tremendous increase in volume of water disturbed by a rolling vessel. For a 15,000-ton vessel at 18 knots—away inside the maximum roll—this loss may easily reach from 1,000 to 1,200 horse-power, and this power is absolutely dissipated and wasted.

Just here the stabilizer steps in with a saving of nearly all of this—practically the entire amount, minus the small and comparatively insignificant quantity of power that is required to keep the gyro wheel spinning in a vacuum. In the course of a very few voyages this power saving in terms of fuel saving amounts to enough to pay for the entire stabilizing equipment.

Very full corroboration of this is found in the service performance of a fast passenger and cargo ship, by taking ten consecutive trips over the same course in the same direction with almost identical load conditions and under conditions of constant propeller revolutions. These trips are sufficiently long, 4,600 miles, to be convincing in the results shown. These data have been plotted in revolutions per knot (see Fig. 9, Plate 44) and give interesting and positive indication of the retardation of the ship owing to weather conditions. A very great amount of the losses in headway due to the retardation effect of the disturbances discussed above will be entirely eliminated by full stabilization. Let us examine what this means in dollars. Suppose the operating expense per 24 hours to be $6,000. The extra expense—that is, the expense over and above the average—in the stormy months amounts to not far from $100,000. This, taken with the amount saved through elimination of bilge keel losses, develops an earning capacity of the stabilizer of not far from 100 per cent per annum. All of this is over and above the many other important gains, both direct and indirect, resulting from the stabilizer installation.

The stabilizer achieves another economy of very great significance to both the operator and the passengers of fast ships. This is the practical avoidance of the necessity for slowing down ships in stormy weather or when heavy seas prevail, the ships being able to make practically the same time under storm conditions. This has been repeatedly demonstrated and is a result so startling that, when first experienced, it has often been claimed as an original discovery by the skippers of stabilized ships.

So insignificant are the stresses required to prevent all rolling that it is interesting to compare them with, or to state them in terms of the specific loading permitted by the underwriters. Take a concrete case of the transport Henderson, of 10,000 tons displacement, 488 feet length, with maximum beam of 48 feet and draught of 19 feet 9 inches—the allowable load per running foot for the section where the stabilizer is located is 14.5 tons. The load due to the weight of the stabilizer plus the maximum gyroscopic stabilizing moments, figured as load upon the vessel, is at maximum only 10 tons, and with average stabilizing moments about 6 tons per running foot. In other words, the stabilizer loads are much less than normal cargo loads. Some of the vessels carrying heavy machinery to France were loaded as high as 28 tons per running foot. The foregoing makes clear, in a very practical way, the relation these forces bear to the vessel's ordinary loads.

Another definite advantage secured by the stabilizer is the elimination of the bilge keels. Dealing only as they do with "V" square, they can never be of service other than in the heaviest rolling—at all other times they are a positive menace. The well-known drag of bilge keels in perfectly calm weather is not only ever present but represents positive losses, even in excess of those calculated. This has now been positively observed in the case of a large 20-knot ship, the performance with and without bilge keels for the same shaft revolutions, loading and trim being known. Moreover, a ship is never trimmed very accurately longitudinally, giving the bilge keels a frontal attack component with the attendant eddies, consuming additional power of no small magnitude. Fig. 10, Plate 45, gives diagrams illustrating this point.

In rough weather the bilge keels afford an extra opportunity for the waves to lay hold of the vessel in rolling. Recently it has been definitely determined that the power required by bilge keels under the condition of pitching, even in a moderate sea, increases the propelling power to a point much beyond what had been supposed. In the case of a 10,000-ton vessel with standard keels, even with moderate pitching, an increased propelling power of about 9 per cent has been observed. In a stabilized ship it is possible to eliminate bilge keels. There should be no hesitancy in omitting keels, as an exceptionally large and successful fast passenger ship has been operated without bilge keels for years.

A reduction in stresses of the propelling machinery of a stabilized ship represents another important gain, especially in the case of twin-screw vessels where the windward propeller is held very much more satisfactorily to its duty—not only saving power but preventing the racking strains due to overspeeding when the screws are "rolled out." The efficiency of a propeller falls off abruptly as its blades even approach the surface, especially where the dip of the waves allows the slightest aeration of the water. For efficiency a propeller requires to be kept down in stiff water.

Prevention of deterioration in cargo applies especially to ships carrying live stock. Figures have been furnished by a concern transporting horses during the war showing that, in a heavy storm during a single trip, their losses per trip often amounted to $30,000 or $40,000, a sum sufficient in a short period to equip the vessel with a stabilizer.

The equipments of more recent design are single gyros and much simplified. These are illustrated in Fig. 11, Plate 45, showing a complete stabilizer plant for a 530-ton yacht assembled in the shop for test; also by Fig. 12, Plate 45, showing a stabilizer for a light cruiser in its shipping frame.

The ability to roll the ship has proved in actual experience to be important in case of emergency to free the vessel from sand and mud banks, by opening the contacting crevices and gradually liquefying the encumbent mass. This has been discussed in a former paper, also the field of the active gyro in rolling ice-breakers and preventing them from freezing in when cutting through rivers and harbors during the winter months. The most important use for rolling, however, is as an aid to gunnery.

As outlined, we have been accumulating a large amount of actual sea experience with various sized equipments. The Government has been of the greatest assistance in encouraging the development. The war has seriously interfered with the work, but even during this period the Navy has allowed us to complete the largest gyro equipment yet attempted, and though the installation has been retarded by the ship being in constant transport service, this fact has offered additional opportunity for final testing. Fig. 13, Plate 46, gives a very good view of the U. S. S. Henderson, a transport built at the Philadelphia Navy Yard. Fig. 14, Plate 46, shows a corner of the gyro room on the Henderson with the precession motor in the foreground, and Fig. 16, Plate 47, shows the little control room with the rotary for generating A. C. current for spinning the gyros in the background. Directly in front of the rotary converter is the vertical gudgeon bearing which transmits the stabilizing moments through the ordinary steel decking to the ship's structure. This ordinary steel decking is found to carry the stabilizing moments with a large factor of safety. In the foreground is the special control gyro for the ordnance tests. In this case the gyro may operate in a single period successively as a stabilizer or as rolling equipment. Fig. 15, Plate 47, shows a curve of the Henderson automatically rolled by her special equipment when at sea under full headway.

This ship has operated with as high as 11,500 tons displacement, and the stabilizer has been repeatedly operated under overload conditions without difficulty, the journals running with perfect temperature control under the heaviest duty, including overload conditions. Even different lubricants have been tried, all giving about the same results; in fact, the plant, since the journals have been worked in, has operated normally in service, and the records show that the guaranteed stabilizing moments have been easily developed.

Fig. 17, Plate 47, shows a plain gyro oil bearing that has operated perfectly. This bearing has received over 100,000 precessions, many under overload conditions. This bearing was removed for photographing. The original tool marks and scraping are plainly visible. Ths oil groove to the left, with its far end partially stopped, and the corresponding diametrical one are the feeding grooves proper; the others are for flushing the revolving shaft surface with cooling oil and, together with the exterior holes, serve to hold the bearing under perfect temperature control as described.

This equipment has undergone additional protracted tests as an ordnance fitting, reaching results in this connection which are extremely interesting; operating in this part of the work in conjunction with the base equipment of our latest fire-control system. The exacting nature of these tests and their severity have contributed in no small measure to another extremely interesting result as follows:—The complete knowledge we now have of the behavior of these equipments and the measured results and records have placed us in a position to guarantee unqualifiedly the stabilization of practically any ship to accurate specification and also the equipment by means of which this important result is secured.

DISCUSSION.

The President:—You have heard this most interesting paper of Mr. Sperry on "Non-rolling Passenger Liners—Observations on a Large Stabilized Ship in Service, Including the Plant and Economies Effected by Stabilization." Are there any comments on the paper?

Mr. C. P. Wetherbee, *Vice-President:*—I should like to ask Mr. Sperry two or three questions, for this is a very interesting matter. As I understand it, in your stabilizing work, you only stabilized the transverse component of the ship's motion. You do not make any attempt to stabilize pitching?

Mr. Sperry:—The relative metacentric heights of any particular ship answers that question. One is 200 or 300 feet. It does not take very much to hold the ship from rolling. It is impossible to hold her against the longitudinal motions.

Mr. Wetherbee:—The great stresses produced on a ship's structure in a seaway, according to my experience, are produced by the ship rushing into head seas at high speeds. I do not see how the stabilizer will help that situation.

Mr. Sperry:—That situation does not exist as much as we imagine it does. When the Navy sent our fourteen ships into the Mediterranean in the winter of 1913–1914, we got into tremendous storms going and coming. We had very accurate pitch and roll recorders on board, and we made a number of records at that time. The astonishing thing in searching those records is the very slight angle which represents the greatest pitch of any one of these battleships.

Mr. Wetherbee:—It may be that the ship does not pitch, but the sea exerts a tremendous force against the ship and produces heavy stresses. You can buckle your decks if you do not slow down.

Mr. Sperry:—We have made better time, several times between different ports, in a storm, than we have in calm weather. That ship might have been under somewhat greater stress during the stormy weather, but it was not under the enormous stresses that were introduced when she rolls and pitches also. That is the best way I can explain the matter.

Mr. Wetherbee:—There is another thing—is it not possible that the ship, when rolling in a rough sea, is disturbing less water than when she is not rolling in the same sea conditions? The rolling ship may actually disturb less water than the stabilized one.

Mr. Sperry:—I think the answer to that is the fact that we were able to make in heavy weather better time with the stabilized ship as against the unstabilized ship, that is to say, on the same course and in the same direction.

Mr. Wetherbee:—The last thing is that this precession motor does require a small movement of the ship before it starts in operation—is not that true? It has to get its impulse from something, and that comes from the small initial motion of the ship—is not that true?

Mr. Sperry:—The precession motor does not depend on the ship at all for its operation. It depends on the little pilot gyro which performs its full function the first tenth of a degree.

Mr. Wetherbee:—It does depend on a small motion of the ship to start the thing?

Mr. Sperry:—It certainly does.

Mr. Wetherbee:—I got the idea from the paper that the stabilizer was able to tell when the roll is coming.

Mr. Sperry:—The additional value of the pilot gyro is that it always enables us to proportion the compass according to the desire—no two waves of the ocean are the same—a wave may come from a given side as much as three times, before anything comes from the other side. This tells us and we can adjust the apparatus accordingly. The gyro starts in to precess from a very slight motion, and from that slight motion the gyro is at that time performing the whole act of stabilizing the ship. The motion starts in and the gyro precesses until it has quenched the motion of the vessel and rolled the contact apart.

Mr. Wetherbee:—I have not had very much experience in connection with the handling of vessels at sea, although I have been at sea considerably, and it always struck me on a passenger ship, for instance, the motion you feel is more the pitching than the rolling.

Mr. Sperry:—I am very glad you brought that question up—it has arisen many times in the last ten years. If anybody is in the bow or away back, it does seem as though the motion is very great. Anywhere in the midship section it is not very apparent, because the degrees there are very slight—2.7 degrees is the highest angle you have.

The British Admiralty years and years ago appointed a commission to find out the occasion of seasickness. Their verdict was it was the sum total of all the motion. Sir William White said if you can suppress the biggest motion you can get rid of the biggest part of the trouble. We have made our contribution along the lines of suppressing the greatest disturbance of the roll.

The President:—Will some other gentleman try his hand? (Laughter.) Part of this discussion reminds the Chair of the small boy who was being reprimanded by his mother for pulling the cat's tail. He promptly responded that he was holding the tail and the cat was doing the pulling. As in most things, the proof of the pudding is in the eating thereof. This apparatus for stabilizing has been put in various vessels and has accomplished some good results. I believe it is to be tried on a more extensive scale in the commercial marine. If it is not a real dividend payer, it will not go very far; that would seem certain.

With respect to naval vessels, of course, anything which decreases the rolling adds very much to the military efficiency of the ship, not only in giving greater effective speed results from the propelling machinery, but very much better results from the standpoint of accuracy of gun fire due to greater steadiness of gun platform.

I noted in the paper very handsome tributes to Sir William White, late of the British Service, and Admiral Taylor, of our own Navy. To all who are familiar with the work of Sir William White no tribute could be too great. He contributed, during his long and distinguished career, a very great deal to the development of naval architecture, and was in harness up to the very end. His death was a great loss to the profession he so greatly adorned. As to Admiral Taylor, fortunately he is still with us, and I trust will continue to be for many years to come. Had he censored this paper, I think it more than likely that a large part would have been cut out, at least that part which related to himself. I am very glad he did not censor it, and I should like to avail myself of the privilege of reinforcing the remarks of the author and noting the debt which naval architects owe to a very modest gentleman, who has perhaps done more than any other living naval architect to promote the advancement and usefulness of a profession of which he is such a shining ornament. (Applause.)

I think Mr. Sperry realizes that his paper has created a great deal of interest, and I congratulate him on standing up to all comers. I feel sure the Society will permit me to express its appreciation of his contribution, with our especial thanks for putting the paper in the form he has done.

Our next paper is entitled "Submarines in General—German Submarines in Particular," by Commander E. S. Land, C. C., U. S. N., Member.

I regret very much that the author of this paper has been detailed for service overseas, and therefore cannot be with us today. It would have added a great deal to your pleasure had he been here, because his most attractive personality would have given even greater force to the remarks contained in the paper which he has prepared for us. In his absence the secretary will very kindly read it.

Secretary Cox:—As Admiral Capps has said, it is very unfortunate that Commander Land is not here, because not only is the paper itself extremely interesting and should be presented by him, or by someone familiar with the subject, but in addition I feel sure that if he were here that many of the members would take the opportunity to ask him questions and learn of his experiences. I will read a few of the salient points in the paper.

Secretary Cox then presented the paper.

SUBMARINES IN GENERAL—GERMAN SUBMARINES IN PARTICULAR.

By Commander E. S. Land, Construction Corps, U. S. N., Member.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

I. SUBMARINES IN GENERAL.

The world is quite "fed up" with submarines, thanks to the recent four and one-half years of terrestrial disturbance. While I hold no brief for the vessel as she was generally used by the Huns, this type of craft has proven beyond a doubt her value as an essential part of a navy. This being true, there are a few matters of interest that may bear repetition.

Primarily, there has been and still is a remarkable amount of misinformation about vessels that operate both on the surface and submerged. The veil of secrecy that has enshrouded submarines from the time of Von Drebbel in 1624 to recent times is in the main responsible for this state of affairs. This has been augmented by the veil of mystery in connection with their operations and further augmented by numerous writers with a vivid imagination and a prolific pen.

Facts being stranger than fiction, also rarer, it is rather a pity that the facts with regard to those most interesting and most complicated men-of-war have not been placed before the public. It appears likely that they will be in due course; in fact, many of the most interesting and remarkable stories of the war pertain to the operation of allied submarines, some accounts of which have recently appeared in British periodicals.

In the second place, there exists at the present time a public prejudice against the submarine which is unjust, for the reason that this prejudice should be against the operators who defiled the laws of war, but not against the vessel.

Any implement of war and many implements of peace can be improperly and illegally used. As a matter of fact, most implements of war were improperly and illegally used by the Huns. Therefore, condemn all instruments of war, Q. E. D. Agreed, provided there is to be no more war—otherwise, no man with a logical mind can agree. The history of the world teaches us the impracticability of eliminating any efficient method of conducting warfare. Any doubting Thomas should look up the history of gunpowder, especially its early use by the French and the British. It is equally impracticable to eliminate the construction of submarines. If there is ever another great war (which God forbid), you will find the naval needs of the belligerents well supplied with submarines.

Owing to the secrecy that has always to a great extent surrounded the design and construction of submarines and to the necessary war-time secrecy of their movements, the public has been led to hold wrong opinions of the efficiency of submarines and the very important part they played during the war just ended.

Most of the reports of value about submarines were suppressed by the allied governments, and very little true information reached the public. In addition to this, the methods of warfare waged by Germany with its submarines have tended to cause an impression in the public mind that the submarine as a weapon of war should be done away with. The part that the submarines of the Allies played during the war places the submarine in a different light and reveals its true value not only as a defensive weapon, but particularly as an offensive weapon. Prior to the war, the opinion was prevalent, even among many intimately connected with submarines, that the submarine type of vessel was essentially a defensive weapon and of little value as an offensive weapon. The facts revealed by the experiences of the war have changed this opinion since the submarine has so efficiently proven its worth as a very powerful offensive weapon. Its ability as a scout was truly marvelous, as it was able to gain a maximum of information at a minimum of expense.

Among the things accomplished by the submarine one fact stands out as particularly noteworthy, namely, that more men-of-war were destroyed by submarines than by any other type of war craft. Likewise the submarines proved their efficiency in being the most effective means for watching the movements of the German fleet; in fact much of the information obtained regarding the activities of German war craft was obtained by submarines on picket and patrol duty; furthermore, few movements of German war craft escaped their vigilance.

During the war, the British maintained a picket line with their submarines inside Heligoland Bight. This proved to be an extremely dangerous post to maintain, and the picket line was later withdrawn outside of Heligoland. The British lost six submarines in six weeks during one period of picket duty inside Heligoland; during the entire war they lost twenty-nine out of fifty-eight of their "E" class of boats. Owing to the very dangerous duty performed, it was considered that the normal life for these picket boats was twelve patrols, yet the "waiting list" at Harwich was the longest; this proves the metal of the British Submarine Navy.

The Germans had about thirty submarines at the beginning of the war; during the four and one-half years following they produced or had in process of construction at least six hundred and fifty more, making a total of approximately six hundred and seventy. Of these, two hundred and three were lost during the war. The number of allied surface craft engaged against the German submarines was in the approximate ratio of 200 to 1.

The German submarines destroyed about 15,000,000 tons of shipping and killed about 16,000 people during the war. During the last three months of the war they destroyed about 1,000,000 tons. It was only in October, 1918, that the construction of allied vessels covered the current losses caused by the German submarines.

After we entered the war some of our submarines were located off the south coast of Ireland and some off the Azores. From that moment on the loss to shipping in these localities very appreciably decreased.

It is a rather well-accepted American adage that "It pays to advertise." The secrecy which has enshrouded submarine design, construction and operation has been one of the most serious drawbacks in regard to the development of submarines not only in this country but throughout the world. While this feature may be necessary in war time, there is a good deal of doubt thrown upon the methods followed by the Allies in connection with this matter. Many authorities doubt the advisability of keeping losses of submarine vessels secret, especially when their destruction was accomplished by ordinary methods of war. The idea was to undermine the morale of the enemy, and opinions vary as to whether this was successfully accomplished. There is no doubt that the morale of the enemy was undermined, but, from the present information available, it appears that the morale of the submarine navy of the enemy was superior to that of the remainder of the naval service.

The value of submarines, in both offensive and defensive operations, has never been fully appreciated except by a very small number of technical people. This statement requires no proof, but if proof is desired, all that is necessary is to read the first three chapters of Admiral Jellico's book, "The Grand Fleet." There is sufficient evidence contained in the first part of this book to convince any quantity of doubting Thomases relative to the value of submarines from a military point of view. If it were not such a serious matter, it would almost approach the ludicrous if one determined the amount of energy that was expended by the Grand Fleet in the early stages of the war, owing to the presence, suspected or actual, of one or more submarines in Scapa Flow or the vicinity thereof. The defenseless condition of these bases has been thoroughly brought out in this book, and the herculean efforts made to make these bases submarine-proof form an interesting chapter of the early stages of the war.

"Great stress has been laid upon the value of patrol services as a defense for capital ships against submarines, but no patrol service, however efficient, can guarantee absolute safety to the heavy ships when they are at sea within the radius of hostile submarines, and it would be futile to deny that the menace of underwater attack has considerably modified fleet strategy. Under present conditions the capital ship, as soon as it leaves harbor, is exposed to a form of attack which, if successful, may cause its complete destruction, and will in any case almost certainly disable it. The vessel may be surrounded by destroyers, but even so it is not wholly safe, for several instances have occurred where the submarine evaded the defending screen and planted her torpedoes in the vitals of the large ship. The capital ship itself is all but helpless against an attack from below. She may be heavily armored and carry a powerful battery of main and secondary guns, but at best she will have only a swiftly moving periscope to fire at, while her invisible antagonist has for target a large area of unarmored side and bottom."

"The performance of our own boats during manœuvres proved beyond all doubt the reality of the submarine menace. Nevertheless, down to the very eve of the war there was a decided tendency in senior naval circles to belittle its gravity. The spirited controversy which arose out of Admiral Sir Percy Scott's perhaps too ardent warning brought clear proof of the deprecatory attitude of many of our flag officers toward an arm which even then had reached a most formidable degree of efficiency."

"It would appear that a different policy must henceforth be pursued if the capital ship is not to be driven from the sea by the swift submersible torpedo-vessel whose appearance cannot long be delayed, and of which the British 'K' class submarine may be considered the forerunner."

In concluding my paper on "Submarine Hulls" presented to the Society in 1917, I ventured two predictions regarding which I now take the womanly satisfaction of saying, "I told you so."

One was that the great preponderance of submarine construction was in the vicinity of 800-ton vessels. This is well borne out by a study of the designs laid down during the war, some of which are still under construction.

The other prediction was that the submarine itself is the best antidote for the submarine. Anyone who will analyze in detail the number of submarines lost in the war will find that, due consideration being given to the relative number of vessels engaged in anti-submarine warfare, the allied submarines accounted for more enemy submarines than any other type of vessel.

II. GERMAN SUBMARINES IN PARTICULAR.

It was the writer's privilege to be detailed as a member of the Naval Allied Armistice Commission which proceeded to Germany in December, 1918, on board the H. M. S. Hercules. It also fell to my lot to be detailed as a member of the American Submarine Inspection Board which inspected German submarines after their surrender at Harwich, England. A casual inspection of about one hundred submarines and a detailed inspection of one vessel of each type were made and reports thereon submitted to the Navy Department. It is, of course, out of the question to cover all the matters with regard to these vessels, but a few of them will be given for the information of the Society.

All modern German submarines are of the double-hull type. The war produced three standard types of German submarines which are known as the UC type, the UB type and the U boat or "mittel U-boat." In addition to the standard types, there were two special types, as follows:—UE type, mine-laying cruisers; UA type, large cruiser class. The large cruiser class consists of two designs, the ordinary design being a vessel of about 2,000 tons surface displacement, while there were a few of the special cruiser design of about 1,200 tons surface displacement. There were apparently only two vessels in this class completed, and they were especially designed for surface speed.

The UC class is made up of small, coastal, mine-laying submarines. This class has a surface displacement of about 500 tons and a submerged displacement of about 575 tons. These vessels in their latest development are about 185 feet long, 18 feet beam, and 12 feet draught. They made about 11½ to 12 knots on the surface and 6 to 7½ knots submerged. The mines were carried in the forward part of the vessel in non-watertight compartments or mine wells, 18 mines being usually carried, although some of the designs carried only 12. A torpedo tube was fitted in the stern and two torpedo tubes were fitted externally in the superstructure, sometimes forward and sometimes abaft the conning-tower. They usually carried one 10.5 or one 8.8-cm. gun. There were about 100 of these boats completed and a great many more in the course of construction.

The UB class of vessels formed the coastal submarines and were vessels of about 520 tons surface displacement and 650 tons submerged. They were 183 feet long, 19 feet beam and 12 feet draught. They had a surface speed of about 13½ knots and a submerged speed of 7½ to 8 knots. Four torpedo tubes were installed in the bow and one in the stern. They usually carried one 10.5-cm. gun; no mines were carried on the UB class.

It is of particular interest to note that the UB class and the UC class in the final analysis were very similar in size and displacement. An examination of the boats of these classes in Germany in December, 1918, showed very completely the progressive steps made in the design and construction of these types of vessels. The earlier UB boats and UC boats only displaced about 125 to 150 tons each. It was very soon demonstrated that these boats were too small for the purpose intended, and when standardization took place it resulted in boats as indicated above. There were about 140 UB boats completed and many more under construction at the time the armistice was signed.

U-boat—"Mittel U-boats."—The first boat of this class was completed in 1906 and had a surface displacement of 238 tons. The increase in the size of this type of submarine continued through various progressive stages until 1914, when the displacement had reached about 760 tons. The "mittel U-boats" subsequent to this date were generally similar in design except that the displacement increased slightly so that the latest type had a surface displacement of about 830 tons. The standard design of "mittel U-boats" has about the following characteristics:—

Surface displacement	830 tons.
Submerged displacement	1,030 tons.
Length	235 feet.
Beam	20½ feet.
Draught	12½ feet.

Designed surface speed, about 16 knots.
Actual surface speed, about 15 knots.
Designed submerged speed, about 8½ knots.
Actual submerged speed, about 8 knots or less.

Most of the "mittel U-boats" were fitted with four bow torpedo tubes and two stern torpedo tubes. They carried either one or two 4.1-inch guns. This type of boat was considered the most successful of the German submarines, and many of the German service consider that, if they had adhered to this type of vessel, they would have been more successful in their submarine warfare. The Germans had completed at the time of the armistice about 110 of these vessels; a large number were also under construction in various parts of Germany.

UE Class.—These were large ocean-going mine-laying cruisers of:—

Surface displacement	about 1,200	tons.
Submerged displacement	1,520	tons.
Length	275	feet.
Beam	24	feet.
Draught	13	feet.

Designed surface speed, 14.7 knots.

Designed submerged speed, 7.2 knots.

Armament, 4 bow torpedo tubes; 2 stern mine tubes, 39 inches in diameter.

Stowage space was available for 42 mines and 24 torpedoes. It is doubtful whether these vessels carried this number of spares. Their gun armament consisted of one 5.9-inch gun and one 4.1-inch, although in some cases they may have carried two 5.9-inch guns. Usually only one gun was carried.

There were about 10 of this class completed and a large number building. U-71-80 were small mine-laying submarines of this general type but of an earlier design, and only displaced about 760 tons. They were not a successful design.

Cruiser Class—UA Type.—Of the special cruiser class there appear to have been completed only 2 boats—U-135 and 136. Their displacement is similar to the mine-laying cruiser class, that is—

Surface displacement	1,200	tons.
Submerged displacement	1,535	tons.
Length	275	feet.
Beam	24½	feet.
Draught	13	feet.
Designed surface speed	18	knots.
Designed submerged speed	8.2	knots.

They had 4 bow tubes and 2 stern tubes and carried one 5.9-inch gun.

The large cruiser class consisted of vessels of the following characteristics:—

Surface displacement	about 2,000	tons.
Submerged displacement	2,500	tons.
Length	302	feet.
Beam	29½	feet.
Draught	17	feet.

They carried 4 bow tubes, 2 stern tubes and 19 torpedoes. They had a designed surface speed of 15.8 knots and a designed submerged speed of 7.7 knots. Only four of these boats were actually completed at the time of the armistice, but there were a large number under construction in various parts of Germany.

A.—It is not the purpose of this paper to attempt to cover the entire question of German submarines, but the following notes on features of the design of these vessels are enumerated:—

These points are selected more on account of minor variations from other standard practices than from their relative importance as regards design.

Frames are numbered from aft forward.

Frames of inner and outer hulls are numbered independently from aft forward, so there are two systems of framing and two sets of numbers.

The hull frames, so far as noted, are the same size above and below the deck level. Stiffening is obtained in way of decks, flats, bulkheads, etc., by means of light brackets at decks and heavier brackets at bulkheads. In a few instances some increases in depth of angle bulb frames to suit local conditions were noted.

The superstructure plating is galvanized, as are also the superstructure frames and beams. The plating varies from 5 pounds to 10 pounds, depending on location; this also varies in different boats.

Three sets of draught figures are secured to the outer hull, one set forward, one set aft, and one set amidships. These are spaced 10 cm. apart.

No battery lining was noted in any boats. The battery tanks are given the ordinary red lead coats, and over this is a bituminous solution or graphite paint—some form of acid-resisting paint. Either they have little or no acid spilled or they flush the tanks with some anti-acid solution, as there was evidence of the acid-proof paint being rubbed off in various places. There was no evidence of acid spilling. The writer was informed at Kiel that they flush out tanks with soda solution.

Various kinds of deck-covering paint were noted on top of the superstructure, including bituminous solution and sand, similar to U. S. Navy efforts on destroyers and submarines.

Only one anchor is fitted. This is a patent deck anchor fitted in a housing hawse pipe on the starboard bow. The shank houses in a tube casting, and the crown and flukes project slightly beyond the ship's side. The hull is recessed to house all but a few inches of crown and flukes. A guard, like a destroyer propeller guard, is fitted around the anchor. The weight of the anchor on the U-124 (1,200-ton boat) is 700 kg. (1,520 pounds), and it appears that the weights of anchors are about 125 pounds per 100 tons of boat. U-71-80 have no deck anchor but one mushroom anchor instead. No other boats of recent type have a mushroom anchor. The mushroom anchor of old UC-boat weighs about 660 pounds.

No armor, as such, is fitted on any of the boats except the U-140, which has a 1-inch armor envelope around the conning-tower. The top strake of the inner hull is 1 inch thick on U-139-142 and ¾ inch on U-135, but only ½ inch to ⅝ inch on U boats and $\frac{7}{16}$ inch to ½ inch on the UB and UC boats. The conning-tower of all late types is very heavy and plating is thick—1¼ inch to 1$\frac{5}{16}$ inch (50 to 52½ pounds). Where the conning-tower dome casting laps the conning-tower plating, a thickness of 2½ inches to 2⅝ inches exists, but this is only a lap; however, it is probably from this that reports of "armor" arose. This plating is a special treatment of bullet-proof steel.

It is a general practice to fit fore-and-aft wood batten decks on the superstructure, following the superstructure contour including gun working circles. These battens are about ½ inch to ¾ inch thick, 4 inches wide, with ½ inch space between battens.

Hand-operating sounding machines are usually fitted in each type, generally located in forward torpedo room in small types or on the third deck between the battery compartment and torpedo compartment on the largest types. This machine is similar to that used on surface craft so far as the principle of design is concerned, but it has a special type of lead with a gauge in it. The machine is operated through a tube casting (gland tube) through the hull, fitted with valves and stuffing box to remove and insert the lead and for watertight purposes. On some vessels the sounding machine is operated by an air motor.

Recognition signal tubes for firing bombs from inside the boat are of recent development and are found on very few boats. The design is quite like our own design and is operated on the same principles. These tubes are usualy located just over the after torpedo room.

All hydroplanes on the latest boats are of the "drowned" type, located well down on the hull forward and aft. They are fixed planes and protected forward by stream-line guards similar to propeller guards on destroyers. Aft they have a half-guard on the after side and a wire rope guard forward or else a complete stream-line guard, similar to forward installation. These guards cut down the submerged and surface speeds very materially—the British claim about one knot for a similar installation on British boats. Cruisers 139 to 142 are exceptions and have above-water forward hydroplanes folding into non-watertight hull. They fold forward and inward, overlapping each other in the stowed position. Guards are fitted above and below the planes. The installation projects slightly beyond the hull lines on either side when folded in.

Two cast-steel or cast-iron pipe wireless masts of the hinged type are fitted. The forward mast hinges aft and the after mast hinges forward in gutters in the superstructure. Some are fitted with steps and a lookout station aloft—usually the forward mast. The thickness of metal at the base is ⅜ inch. These masts are fitted at the heel to a quadrant over which reeves a chain full which raises and lowers them; they are electrically operated from inside the boat. Each section of the mast is swaged or welded together, each section being reduced in diameter from the deck up. There is evidence of the use of acetylene welding in some sections and electric welding in others. They vary in length from 28 feet to 54 feet; average height about 35 feet. It is understood that these masts were seldom used in any kind of a heavy seaway, as it was dangerous to hoist masts or leave them hoisted.

Practically all boats have clearing lines which are insulated and used for low power wireless. These lines reeve from forward to the bridge and aft through a sheave and back to the bridge, where they are fitted with turnbuckles and secured. Some are secured aft, but all are fitted with turnbuckles.

On a number of latest boats were noted an assortment of various sizes of tapered wooden plugs to be used as shot-hole leak stoppers.

The mattresses are stuffed with coarse horse-hair.

The bunks for crew are of the hinged type—pipe rail bunks, with wire springs with pipe bracket at the head end and are fitted with wire springs for pillow. Horse-hair mattresses about 3 inches thick are installed on the springs. The bunks are usually 6 feet long by 23 inches wide.

The aeroplane recognition signal on the forward superstructure deck is a sheet metal ring hinged at the center and can be folded over double when not wanted. The diameter is 6 feet 6 inches and the width of the sheet metal ring is 14 inches. When open, the ring shows white throughout the circle, *i. e.*, 14 inches white painted circle. When closed or folded, the color is the same as the deck.

Very little effort was made to camouflage the submarines, although a few showed signs of having tried various designs—apparently up to whims of the commanding officer. The hulls were generally painted with light slate color and the upper decks or "top side" with black asphaltum paint. A small number of grotesque efforts a la Chinese junk style of eyes on the bow, etc., were noted.

There were usually three external emergency air connections, consisting of two lines each. These were located forward, amidships and aft in the superstructure. The location is marked by a red cross painted near the lines. One line leads to the compartments and one to the tanks. The air lines are 1½ inches outside diameter and are closed by a butterfly cap which screws on the pipe over the outboard end.

Each boat is fitted with two marker buoys—one forward and one aft in the superstructure—which are released from inside the boat. These buoys are fitted with ⅜-inch diameter wire cable. Most of them have no light and no telephone.

The inner hull is built on a flush plating system with double-edge strips and single, internal butt straps.

The outer hull is sometimes built on a raised and sunken system but generally with a flush exterior, obtained by joggling sunken plates, both edges and butts.

Free flooding bow buoyancy tanks are installed in the superstructure—one or two vents operable from the torpedo room or from the control room, or both.

No housing guns are fitted. This practice was abandoned two years ago.

The net cutter bow is knife edge or saw tooth, slanting up and aft at about 30 degrees from the stem-head, and is supported by steel bracket stays of "saw-horse" design. The saw edges are hardened steel insert teeth or knives (10 or 12 of them) about ¾ inch wide, 1 inch to 1½ inches deep and 10 inches long, set between narrow steel holding plates.

Vertical rudders, on some cruisers still building, are fitted above the hull to assist in maneuvering submerged. They are like our G-4 installation. The upper section of the rudder is on the same shaft as the lower section. This practice is not standard even on cruisers.

B.—Of the relatively more important features the following points may be briefly mentioned:—

(*a*) Conning-tower control.

(*b*) Complete double system (supply and exhaust) ventilation.

(*c*) No broadside tubes. One or two stern tubes.

(*d*) Very large heavy conning-towers—of nickel steel when nickel is available.

(*e*) Bulkhead doors in pressure bulkheads are round and very strong.

(*f*) Double hull affords much protection from damage especially by depth charges.

(*g*) Hull strongly built but very wide frame spacing.

(*h*) Heavy concave-convex pressure bulkheads in U, UE. and UB. types.

(*i*) Latest UB. boats have control room fitted with pressure bulkheads.

(*j*) Auxiliaries—all electrically driven.

(*k*) "Drowned" bow hydroplanes except in U-140 class.

(*l*) Chariot bridges fitted in all types.

(*m*) Superstructure deck fitted with a superabundance of ammunition lockers.

(*n*) Many gun emplacements (two to four) but usually only one gun mounted except in cruisers and Deutschland class.

(*o*) Time of diving reduced by "Twin Kingstons."

(*p*) Kingston valve control scattered throughout the boat.

(*q*) Tanks controlled by vent domes and their valves controlled by shafting led into control room.

(*r*) Control of boat in all its operations is divided between conning-tower and control room.

(*s*) Navigation, torpedoes, etc., controlled from conning-tower. Diving, tanks, etc., controlled from control room.

(*t*) All periscopes motor operated. House in steel tubes which extend to keel in some cases. Housing distances 12 feet to 14 feet.

(*u*) After periscope houses in a steel tube or well large enough to take a platform and man up and down.

(*v*) Other periscopes only usable at full extension.

(*w*) Third periscope offset from C.L. with eyepieces in control room. Houses in steel tube, forward and to starboard of conning-tower. Usually have a large, heavy fairwater casting. Very rugged and heavy.

(*x*) Head sections of all periscopes very small, some going down to 1 inch; the latter not believed to be satisfactory. Reported unsatisfactory. Ten feet horizontal base range finder in cruisers.

(*y*) Special periscope attachment permitting fixed seat in conning-tower and yet see through at all heights, thus doing away with platform. This is only on a few of latest boats.

(*z*) Galley is electric—electric cookers used, not electric range.

(*aa*) Air bottles—mostly stowed in superstructure.

C.—The outstanding important features of these boats are:—Excellent engines, excellent periscopes, double-hull protection, large surface radius of action.

D.—A few of the objectionable features and defects may be briefly enumerated:—

1. No effort made to insulate auxiliaries.
2. Apparently little effort made to eliminate noise of auxiliaries.
3. Elaborate ventilation and cross connections.
4. Elaborate air purifying installations. Altogether too elaborate.
5. Superabundance of shafting running from control room.
6. Bulkheads penetrated by myriads of leads of all kinds.
7. Only one hatch in UC. conning-tower—an element of weakness.
8. Batteries are inaccessible except in UE. and UA. classes.
9. Folding wireless masts—heavy, awkward, and cumbersome installation.
10. Many emergency leads and connections of all kinds. Altogether too many.
11. Much space given up to officers' quarters and chief petty officers; not much space or comfort for crews.
12. Little attention paid to hull excrescences.
13. Submerged speed considered of minor importance.
14. Boats are not well designed as to exterior of hull, so far as speed is concerned, either surface or submerged.
15. With the horse-power the engines give, greater speeds should and could be obtained.
16. Boats are crowded, congested, complicated and filled with "gadgets" of all kinds.
17. There is a good deal of doubt about the stability of the mine-laying cruisers, and the large cruisers, particularly submerged stability. Ample evidence was obtained to show that the Germans considered this a very serious question with regard to these boats and various modifications and alterations were under way to improve the stability of these designs.

It is quite apparent that the trend of the German submarine construction was to give the U. S. A. a thorough taste of "Frightfulness" in the spring of 1919.

DISCUSSION.

The President:—Gentlemen, Commander Land's paper entitled "Submarines in General—German Submarines in Particular," is before you for discussion, and I trust that someone will take up the discussion. It may be a little helpful to you to have the Chair refer to some ancient history. In coming to New York on the train a few days ago I began to glance over the list of the papers read before this Society—not a very exacting mental performance, but calculated to while away the time profitably—and I ran across this very interesting statement of a "topic for discussion" at our annual meeting in 1898, twenty-one years ago—"The Utility of Torpedo Boats, and Has the Submarine Boat a Place?" That subject for topical discussion, read today, is most interesting. The fact that ships aggregating 15,000,000 in tonnage were sunk by submarines during the years of grace, 1917–1918, is a sufficient answer to the question—"Has the submarine a place?" if some nation has submarines and chooses to use them.

The other subject attached to this topical discussion of twenty-one years ago is equally interesting. It was "The Utility of Torpedo Boats." The torpedo boat has long since grown into the torpedo-boat destroyer, and the vessel which, more than all others, threw the fear of the Lord into the enemy submarine fleet was the destroyer, because of its numbers, its power to maneuver quickly, its comparatively shallow draught and high speed, and, above all, its ability to drop promptly and in the right place high explosive depth charges which, at a comparatively small distance, would do a great deal of damage. From this we can see that our Proceedings have not only a great deal of technical value but a good deal of historical interest besides.

I trust that someone will make some contribution to the discussion. This is a paper that has a great deal in it, of course, of especial interest to the naval architect, but also has interest for all of us as ship designers. One is almost tempted to put the direct question to the last gentleman who entered the room. We are trying to get up some discussion on a very admirable paper on submarines. We know of a certain company that did a great deal in the construction of submarines.

Mr. J. W. Powell (the gentleman referred to by the chairman):—I suppose on the theory that I have not read the paper, I am particularly well qualified to discuss it?

The President:—Well qualified——

Mr. Powell:—The submarine, of course, has undergone a very rapid and a very remarkable development, and a good many of us think it is still going to play a very, very considerable part in future warfare, provided the League of Nations does not do away entirely with the necessity of any warlike vessels. My own personal feeling is that we are probably getting very near the day of the end of the big surface ship, and the development of the submarine is going to more and more change the form of the surface warfare from what it has been in the past. I do not know how these remarks gee up with what the paper says, but that is the feeling which has grown in my mind as I have watched the development of the submarine.

The President:—We owe our thanks also to Mr. Powell for so quickly responding to the invitation to speak; if no one else cares to discuss the paper further, the Chair feels he will be entirely expressing the feeling of the meeting in conveying its thanks to Commander Land for his admirable paper. Again, I wish to say that I regret very deeply that he is not here in person, as I feel quite sure that he would have thrown a little ginger into the after events.

Chief Constructor A. W. Johns, C. B. E. (Communicated):—Commander Land's paper is an interesting one and reflects the virile personality of its writer. He points out that from two main causes—secrecy and method of use by the Germans—the submarine has a most evil reputation. This is undeserved, and those who study its design, construction and operation will find it a most fascinating subject. So far as design is concerned, for its surface navigation the same qualities are necessary as for surface ships. For submerged navigation a new set of circumstances arise which must be superimposed on these already mentioned. Design is therefore far more complicated and difficult than is the case for ordinary ships. Its small size and the multiplicity of its fittings render construction and completion also more difficult. So far as its method of operation during war is concerned it is unpopular, because, as many point out, it skulks below the surface and waits and watches for its prey. They forget it is only utilizing one of the most ancient maxims in war and sport; *i. e.*, that of taking cover. It does this in the only effective manner possible in sea warfare.

There is, however, another reason for some of its unpopularity. It has to be remembered that "submarining" is a young man's game and only a relatively small number of naval officers serve in the submarine service. Its introduction is relatively recent and, consequently, it is only a very few of the senior officers of the services who have any personal knowledge of its value and limitations. Naturally the older senior officers considered it a new-fangled contrivance and looked upon its advent with suspicion, if not with disfavor. Its employment in warfare fundamentally altered many of those principles of tactics and strategy with which they were familiar. A privileged few of the senior officers were taken out on a submarine on show occasions, saw how smoothly its operations were carried out, and went home and wrote letters to the press on the futility of building battleships. They had seen it at its best and could not appreciate its limitations. At the present time this unpopularity is decreasing, thanks to the lessons of the war, and also by the entry into the senior ranks of the service of some of the earliest and most experienced submarine officers.

Luckily for the Allies, Germany's naval autocrat, Von Tirpitz, one of the senior officers, was not an enthusiastic admirer of the submarine, and consequently Germany built few before the war, starting their construction far later than other nations and apparently only then because the latter were increasing their numbers.

Commander Land points out that the submarine has proved beyond a doubt its value as an essential part of a navy. Nothing has confirmed this more than the reception given to the semiofficial announcement that the British Admiralty were in favor of its abolition. It might be accepted that a nation which has a large surface fleet of war and merchant ships, and which suffered more serious losses during the war than any other nation from the submarine, would be favorable to its abolition. The public press stated that practically all continental nations had strongly objected to the proposed abolition and, as far as can be judged, the matter has dropped.

In the paper it is pointed out that the majority of boats built during the war were of about 850 tons displacement. Size is generally dependent on the weight and power of the engines available at the time of the design, as the latter has to be built around them. Most of the nations have developed submarine engines of from 1,000 to 1,200 B.H.P. as a maximum and it almost naturally follows that the tonnage displacement of the boats built with these engines should be about the same. As soon, however, as the Germans had more powerful engines available the size of the boat showed an increase. Thus the 1,750 B.H.P. engines, fitted in U135 and U136, required a displacement of 1,150 tons, and the 3,000 B.H.P. engines intended to have been fitted in U142 increased the displacement to 2,150 tons. All other classes of warships have increased in displacement, speed and other qualities as soon as progress in marine engineering rendered more powerful engines available, and in this respect submarines will undoubtedly evolve along similar lines.

It was my very pleasant duty to accompany Commander Land in his inspection of German submarines, and I can confirm all that he has written of them. They were well-built boats but filled with "gilguys" and "gadgets" which are not usually found in the submarines of other nations. It was surprising to find many unnecessary fittings which must have taken time and labor to make and install, and at a time when celerity of completion was of the utmost importance to the Germans. Nothing in the boats bore the mark of being constructed or fitted hastily. Stories of rapid building reaches the allied countries, but information since received shows these stories to be false and that building and completion were slow. This is also confirmed by the cost of the boats, which is relatively much greater per ton than in other countries.

The 350 boats actually completed during the war cost about fourteen hundred million marks. The value of the losses to allied merchant shipping and cargoes, and also the cost of anti-submarine measures, have not yet been computed, but must represent an immense sum—many times greater than the cost of the submarines.

In conclusion I wish to express my most cordial appreciation both of the paper and of my most amiable and pleasant colleague on many journeys and inspections—the writer.

Commander Land (Communicated):—It is a great regret to me that I was unable to be present at the meetings of the N. A. and M. E., not only to present my paper but also to hear the other papers presented to this Society.

Submarines, with us, have been such a restricted proposition that it is difficult to arouse much interest or much discussion before this Society. I very much appreciate the kind remarks of the chairman, who has done so much, not only for the advancement of submarines and submarine design, but also for the advancement of all warship designs for the United States Navy.

My own convictions are to the effect that we have never taken the question of submarines for our service with sufficient seriousness to bring about the best results. There is something inherent in this type of war vessel which does not appeal to the sporting instincts of the *genus homo;* there are also many things about submarines which do not appeal to the older officers of the service, these points being quite well known to anyone who has given the matter serious consideration. The submarine service, therefore, resolves itself into a "young man's game." There is no lack of enthusiasm in the younger elements of the service, and "submarining" requires, at all times, the best that is in a man.

In this connection I cannot refrain from quoting a remark I have heard over here several times which emanates from the British Naval Service:—"This would have been a jolly little naval war if it had not been for submarines."

Writing from foreign soil (England), I am more than ever convinced of the importance of the submarine branch of the service in case we are ever again involved in war. As the members of this Society are undoubtedly aware, there has been considerable controversy in Great Britain over the advantages and disadvantages of submarine craft and aircraft as against the surface ship. While this newspaper discussion does not carry conviction for either side, it certainly indicates that for a well-balanced fleet the development of submarines must be most carefully considered and progress must be made in order to accomplish results which will be absolutely necessary for the naval defense of our country. I therefore wish to impress upon the members of the N. A. and M. E. the vital importance of "carrying on" the development of this very important arm of the Navy.

The President:—We have now come, gentlemen, to the end of our day's program. Tomorrow morning at 10 o'clock we will reconvene and proceed to the second day's business. I need not remind you again that the banquet is on Saturday night at 7 o'clock, and that the boat for the excursion in New York Harbor leaves at 11 o'clock Saturday morning, and that all members and their guests and the ladies of their party are invited to be present. The vessel will land at the shipyard of the Submarine Boat Corporation, and the party will there witness a launching. Luncheon will be served on board the Chester W. Chapin.

The meeting will now stand adjourned until Friday morning at 10 o'clock.

THIRD SESSION.

Friday Morning, November 14, 1919.

President Capps called the meeting to order at 10.15 o'clock.

The President:—The meeting will please come to order. We will begin the proceedings of the day by taking the next paper in regular order which is entitled "Propulsive Efficiency of Single-Screw Cargo Ships," by Commander Wm. McEntee, C. C., U. S. N., Member. In the absence of the author, Professor Sadler has kindly consented to present the paper.

Professor Sadler presented the paper.

THE PROPULSIVE EFFICIENCY OF SINGLE-SCREW CARGO SHIPS.

By Commander William McEntee, Construction Corps, U. S. Navy, Member.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

In a paper* read before the last meeting of the Society giving the results of a series of experiments made at the U. S. Experimental Model Basin with models of single-screw cargo ships it was shown among other things that for good propulsive efficiency it was necessary to have the run finer than the entrance, and for vessels of this class having a parallel middle body equal to about one-third the length of the ship the minimum power is required when the fore-and-aft position of the parallel middle body is such that about two-thirds of it is forward of the midship section and one-third abaft it. Those results, which it is believed were the first of any extensive series of experiments with self-propelled models of single-screw ships to be made public, were confirmed by a somewhat similar series of tests made at Clydebank on models of vessels of the same general class. The results of the latter tests were presented in a paper† by Mr. James Semple before the Institution of Naval Architects in April, 1919.

In view of the importance of this class of vessel it was considered desirable to extend the investigation, and the present paper contains the results of the tests of five additional models, the characteristics of which are given in the following table:—

Table I.

Model number	Longitudinal coefficient	Per cent of parallel middle body	Run in per cent of length	Displacement—tons
2181	0.74	30	38.70	12.341
2182	0.76	35	37.03	12,675
2183	0.78	40	35.37	13,009
2184	0.80	45	33.70	13,344
2185	0.82	50	32.03	13,678

The lines of the parent form, Model No. 2023, are shown in Plate 48. In Plate 49 are shown the sectional area curves for the five models and the parent form.

The investigation had for its object determination of the effect of varying the amount of parallel middle body and the longitudinal coefficient simultaneously while

*"Variation of Shaft Horse-Power, Propeller Revolutions and Propulsive Coefficient with Longitudinal Position of the Parallel Middle Body in a Single-Screw Cargo Ship."

†"Experiments on Full Cargo Ship Models."

maintaining the principal over-all dimensions. In other words, the investigation sought to obtain data from which it would be possible at any given practicable speed for a cargo carrier to determine the economic possibilities of increasing the displacement by increasing the parallel middle body and longitudinal coefficient. A great many vessels of this character now built have coefficients ranging from 0.78 to 0.80, and the question to be determined was the economic limit of fullness so far as coefficient and displacement are concerned for a vessel of a given length, beam, and draught. In the five models tested the longitudinal coefficient varied by increments of 0.02 from 0.74 to 0.82.

As mentioned in last year's paper, previous experiments at the Experimental Model Basin have shown that, for a single-screw cargo ship of about 0.78 coefficient, it is desirable to use a parallel middle body of about one-third the length. For simplicity in construction, however, it is desirable to use a somewhat greater length of parallel middle body. It is possible to use with this coefficient a parallel middle body of about 40 per cent of the ship's length without much sacrifice in power. This consideration determined the extent of parallel middle body for each model as given in Table I. Comparison of results of the present series with last year's results shows that an increase of parallel middle body from 33 per cent to 42 per cent causes an increase in estimated horse-power of but 3 per cent only.

In order to keep the run fine, the fore-and-aft distribution of the parallel middle body was retained the same as had previously shown the best results; that is, one-third abaft the midship section and two-thirds forward of it. This distribution possibly gives for the higher coefficient a model which is too bluff at the entrance, especially at the higher speeds. For the lower speeds, the distribution of sectional area seems to be satisfactory, particularly as regards eddying at the stern, which is liable to occur with vessels of full coefficient.

The following description of the dynamometer and the methods of making the tests are repeated from last year's paper for the benefit of those who may not have a copy at hand.

The models were carefully made and all were fitted with the same cast stern frame, which included the stern bearing for the propeller shaft. The stern frame had the rudder cast with it. The whole frame and rudder was fitted to each of the four models before the self-propulsion experiments were undertaken, and, together with the propeller shaft, propeller, and dynamometer, was transferred from one model to the other as the experiments with each model were completed.

The dynamometer consisted of a small direct-current motor, the armature shaft of which was directly connected with the propeller shaft by means of a flexible coupling. The armature shaft was free to float fore and aft in its bearings about $\frac{7}{16}$ inch in an axial direction. The armature shaft was connected to a calibrated spring by means of a thrust bearing, so that the axial displacement of the armature shaft gave a measurement of the propeller thrusts. Similarly, the frame of the motor was mounted so as to rotate in independent bearings. The torque developed by the motor acted against a calibrated spring so that the deflection of the spring indicated the torque of the motor. In addition to this there were suitable means provided for measuring the revolutions of the shaft.

The order of procedure in making the tests was as follows:—The shaft and dynamometer were carefully lined up and the whole run for a sufficient time to warm up the bearings and reduce the bearing friction as much as possible. Owing to the fact that the dynamometer was placed very close to the stern, but a short length of propeller shafting was necessary, and this was supported by two self-aligning bearings, one at the stern bearing and the other at the forward end of the stern tube. With the propeller shaft in place and everything working freely, the model was towed in the Model Basin beneath the towing carriage at several different speeds, and the propeller shaft, without propeller, run at the range of revolutions to be covered in the course of the experiments. The propeller was then fitted to the shaft, and cards for torque and thrust and revolutions per minute were taken with the model self-propelled at different speeds. In these tests the model was guided by two plates about 10 inches in width placed at either end of the model so as to steer it in a straight course. The guide plates floated between the guiding points attached to the carriage, but the towing carriage did not exercise any force on the model in a fore-and-aft direction. Starting at low speeds corresponding to about 5 knots for the ship, the towing carriage was adjusted to run at a uniform speed. The rheostat controlling the speed of the propeller dynamometer was then adjusted so that the thrust of the propeller would just keep the model running as fast as the towing carriage, without striking the stops, which were placed at an interval of 6 inches. Thus, starting with the model in the mid position, it was free to gain or lose a distance of 3 inches as compared with the towing carriage before striking either stop. When the propeller was running at the proper speed to keep the model up with the towing carriage, the record of thrust, torque and revolutions per minute was taken. If, in the course of the run, the model struck either stop on the carriage, the run was discarded and another run made. Having obtained the desired data at the lowest speed, the carriage speed was increased for subsequent runs and similar data taken at higher speeds. The range covered corresponded to speeds of 5 to 13 knots for the ship. About forty different runs were made with each model, giving a corresponding number of points for plotting the torque, thrust and revolutions per minute curves.

The armature of the propeller dynamometer was especially designed to reduce to a minimum the amount of magnetic thrust. This thrust increased with the torque and amounted to 0.17 pound when the armature was displaced $\frac{7}{16}$ inch and the torque delivered to the shaft was 16 pound-inches. Neglecting this at higher powers would have caused an error in thrust measurements of about 1.4 per cent, but would not have caused any error in the power measurements. However, this magnetic thrust was separately calibrated, and corrections for it were made in working up the results of the experiments.

Immediately after completion of the self-propulsion tests on the model the propeller was removed, and the runs to obtain the shaft friction and the thrust without propeller were repeated. The model was next connected with the resistance dynamometer on the towing carriage and the usual model resistance data taken. This insured that the conditions of test, both for self-propulsion and for the resistance of the model, would be uniform as regards conditions of the model, temperature of water, etc.

The following are the dimensions of the propeller used in the experiments and also the dimensions expanded to the ship scale:—

	Model.	*Ship.*
Diameter	10.125 inches	16 feet 7 inches
Pitch	9.0 inches	14 feet 9 inches

Pitch ratio	0.889
Mean width ratio	0.20
Number of blades	3
Ratio of projected to disc area	0.266
Blade thickness fraction	0.04

The propeller had three blades of Taylor's standard form.

The propeller characteristics were obtained by separate tests of the propeller model run in free water, that is, in a separate apparatus where the propeller shaft projected well ahead, so that the propeller ran in water undisturbed by the action of the testing apparatus. The same motor dynamometer was used for tests as was used for the self-propulsion tests, the only difference being that the propeller shaft was coupled to the forward end of the armature shaft instead of the after end.

The characteristics of the propeller are given in Plate 50. The thrust constant, C_T, and the torque constant, C_Q, are plotted on nominal slip following the method used by Schaffran.* These constants, which are in non-dimensional form, lend themselves well to the analysis of self-propulsion experiments and to the extension of the results to the full-sized ship.

The results of the investigation are given in Plates 51, 52 and 53. An examination of the estimated horse-power curves and the shaft horse-power curves for the various models shows, as was to be expected, wide variation in power requirements, especially at higher speeds. For example, at 11 knots passing from Model 2181 longitudinal coefficient 0.74 to Model 2185 longitudinal coefficient 0.82, with a change in displacement of 1,337 tons, or 10.8 per cent, increases the effective horse-power from 1,165 to 1,585 or 36 per cent, while the shaft horse-power is increased by 950 or 54 per cent. In this range the percentage increase in effective horse-power required is about three and one-half times as much as the increase in displacement, while the increase in shaft horse-power is five times as great as that in displacement. At a speed of 11 knots the increased displacement is obtained by an apparently unwarranted increase in power. This results from the fact that the

*"Systematische Propellerversuche"; K. Schaffran, Schiffbau, September 22, 1915.

natural increase in effective horse-power resulting from the larger displacement and coefficient is magnified or augmented by an accompanying fall in the propulsive coefficient from 0.665 to 0.587.

In the tests the same propeller was used on all models, and it may be considered that the propeller is too small for the ship represented by Model 2185, as the revolutions per minute have increased from 84.5 to 93.2. This is to some extent true, but separate estimates indicate that if a propeller of different diameter or pitch were fitted to 2185 for the purpose of reducing the revolutions to 84.5, such change would make but little change in the propulsive coefficient, not over one per cent, so for all practical purposes the models may be compared on the basis of the shaft horse-power curves shown in Plate 52 without taking the revolutions into consideration.

As another comparison of these results, it is interesting to consider the effect of increasing the longitudinal coefficient and displacement of the ship represented by Model 2183. The curves show that at a speed of 11 knots an increase of longitudinal coefficient from 0.78 to 0.80 with a corresponding increase in displacement of 335 tons or 2.57 per cent requires an 8 per cent increase in effective horse-power and 12.27 per cent increase in shaft horse-power. Of the increased displacement, only a part would be available as cargo-carrying capacity, depending upon the necessary increase in the weight of the hull, machinery, and fuel. On a given trade route, with a knowledge as to the cost of fuel and other expenses proportional to the power, it may readily be determined whether the additional cargo carried for the additional displacement would be economically possible with the increased power charges.

In Plate 53 are shown the curves of wake fraction, thrust deduction coefficient, apparent slip and true slip for the ships. In extending the results of the model experiments to the full-sized ships it has been assumed that the wake fraction and thrust deduction coefficient for the ships are the same as for the models.

As the American practice in defining the wake as a percentage of the ship's speed varies from that followed in Great Britain, the following definitions of the thrust deduction coefficient and the wake fraction are given:—

$$t = \frac{T - R}{T};\ w = \frac{V - V'}{V},$$

in which T is the thrust of the propeller, R the resistance of the ship, V the speed of the ship, V' the speed of advance of the propeller in the water in which it works.

With regard to the shaft horse-power required for the ships represented by the different models, it should be noted that the curves show what may be expected under trial conditions, that is, with a clean, freshly painted bottom and a smooth bronze propeller running in smooth water. To maintain the same sea speed over considerable periods, it is necessary to allow a margin of power to cover the increased resistance due to average sea and weather conditions and a moderate amount of fouling of bottom which occurs between each docking and painting.

Summing up the results of both series of experiments, it appears safe to conclude that, for a well-designed hull with a propeller running at about 90 revolutions per minute, a hull efficiency of 1.09 and a propulsive efficiency of 0.65 may be expected in a single-screw cargo ship.

DISCUSSION.

THE PRESIDENT:—Commander McEntee's paper on "Propulsive Efficiency of Single-Screw Cargo Ships" is now before you for discussion, gentlemen.

MR. A. J. C. ROBERTSON, *Member:*—I wish that Mr. McEntee were present here so that we could express to him our great gratitude for this paper and the paper of last year, the two really constituting one paper. Mr. McEntee has worked out a system of self-propulsion tests which have given us very valuable results. They are presented in last year's paper and this paper. In last year's paper Mr. McEntee took a single prismatic coefficient for his model of about 80 per cent. He took a fixed parallel middle body and moved it by steps from aft of midship toward the fore body of the ship and gave us the resistance curves and the shaft horse-power that resulted, and if anyone takes the trouble to plot the curves of last year, he will find that a very full bow with a very fine stern for speeds around 6, 6.5, 7 and up to 8 knots gives distinctly the best form. When the speeds go above 8 knots the formation of waves and the resistance due to the wave-making at the bow evidently counter-balance the gain of a fine stern in the wake and eddy-making aft, so that the proportion which is best for driving gradually moves the middle body aft. However, the fine stern Mr. McEntee secured by putting the parallel body very far forward gave very high propeller efficiency, and for that reason he selected a model with a parallel body placed considerably further forward than it should be for the best power resistance results.

In this paper Mr. McEntee refers to the change between last year's model and this year's model, showing an increase in resistance for this year's model of 3 per cent. He does not state that that 3 per cent is at 11 knots, which is the designer's speed of the vessel. At 12 knots the increase of this model over the best model of last year is more than 11 per cent, which shows that the fullness forward which he advocates in order to secure a good propeller efficiency would have a serious effect if the vessel were going to meet a head sea, which would augment its resistance over the resistance at 11 knots.

There is one point further in regard to Mr. McEntee's figures I wish to refer to. The 65 per cent propulsive efficiency which he states is obtained with the prismatic coefficient of the after body of the ship of only 70 per cent, and to increase the after body prismatic to 74 per cent, cuts down the coefficient of propulsion by 4 per cent, bringing it down to 61 per cent. Now a vessel around about 80 per cent prismatic coefficient would have its center of buoyancy extremely far forward if it had to have an afterbody prismatic of only 70 per cent, so that is a qualification of the use of this very high efficiency which Mr. McEntee foretells.

There is another qualification I want to make in regard to that figure. Suppose we take a ship proportioned such as we would like to send to sea, proportions in which we can locate the machinery where we want it, that would bring us down to about 61 per cent propulsive coefficient, and that is comparing the shaft horse-power with the engine horse-power of the model which was without bilge keel. If we add 5 per cent to that effective horse-power to cover bilge keels and wind resistance, and if we take the efficiency of propulsion of the reciprocating engine, which would normally be compared with this, at 90 per cent, the coefficient of propulsion, instead of being 65 per cent, would only be about 52 per cent, which is what good practice seems to indicate today.

The President:—Is there anyone else who desires to speak on this paper?

Mr. W. L. R. Emmet, *Member of Council:*—I have no expert knowledge of the subject-matter of this paper, but there is something which has recently been observed which seems to have an interesting bearing on the information contained in this paper.

The General Electric Company's men have recently built and put on a ship, on a tanker of about the character of this ship described here, a torsion device which connects the shaft to the propeller. This torsional device is arranged with springs and admits of considerable torsional movement. To this torsional device we attached a train of connections so that we could get a stationary record from the propeller under all conditions of running with a view to seeing what the variations of torque in the shaft were. This shaft was being run by a turbine, and the fly-wheel effect of that turbine was known. While we did not measure the speed variations of the turbine, we knew what they were by the variation of torque.

We sent a man to sea in the vessel. Unfortunately we did not get any very rough weather, but got some moderately rough weather, four degrees of pitching, and it showed an enormous variation of torque in the propeller—that is, the torque went from zero to 75 per cent above normal from pitching of the ship.

Furthermore, in this ship, when this was happening the propeller was not lifted out of the water. It was constantly submerged, so that the variation of torque was purely a matter of the relative speed of water and propeller. Furthermore, the observation, which is being verified by actual automatic records now, showed that the loss of torque was not incident to the lifting of the propeller to the surface, to cavitation, or anything of that sort; it was thought to have happened when the stern of the ship was settling into the water, so that the water was being thrown aft by the shape of the stern.

There is no doubt in my mind that this is what has been destroying all the gears and unquestionably the conditions are infinitely worse under certain combinations of sea, because these peculiar results only occasionally occur, and certain combinations of waves would produce two or three periods of very heavy strain.

If we compare the facts shown by Mr. McEntee's paper with the indications which I have stated, we will see that the speed relation of water to the propeller must be subject to a large variation in order to create such changes of torque. The actual slip given by Mr. McEntee is very large, and this slip must be overcome at one time and greatly increased at another.

Such variations cannot fail to make a tremendous difference in the efficiency of the propeller, and I am inclined to think, in the case of a ship in even a moderate seaway, there would be greater reason for using such characteristics as Mr. McEntee recommends than in the case of a ship in smooth water—that is, the stern profoundly affects the action of the propeller, but will affect it more in a seaway than in smooth water, and consequently, in order to decide what sort of a propeller a ship should have, it would presumably be necessary to investigate what the propeller is doing in rough water.

Then there is another matter in that connection which struck me as interesting, namely, that this effect would be greatest in a relatively low-speed propeller, with large area and small slip, and these variations might be a reason for using smaller higher speed propellers which might, in smooth water, be less efficient, but which might in rough water show a better result, as they would not be so greatly affected bv slight variations of the relative speed of the water.

Professor H. C. Sadler, *Member of Council:*—I would confirm what Mr. Robertson said about propulsive coefficients. We have found with the ordinary tank tests and bare hull that the ratio of effective to indicated horse-power in practice works out between 50 and 52 per cent—52 per cent with a good machinery installation. If you have reason to suppose that ordinary care has not been given to the installation of the machinery, it is a little safer to take 50 per cent. I mention this because the figure of 65 per cent given in the paper may be a little misleading, if members feel that is the ratio of indicated horse-power to effective.

With regard to the question of trial trip speed and average sea speed, it has been my experience that it is safe to say that with the same horse-power the reduction in speed over a year's working in a vessel amounts to very nearly 10 per cent—somewhere between 8 and 10 per cent—so that if you want to figure on sustained sea speeds, and have the trial trip result, I think you are fairly safe in saying that the speed will be reduced somewhere about 10 per cent.

Mr. F. M. Hiatt, *Member:*—In speaking of the propulsive coefficient, are you referring to the ratio of the resistance, with appendages, or the bare hull resistance, plain model resistance?

Commander William McEntee, C. C., U. S. N. (Communicated):—Referring to Mr. Robertson's discussion as to the relative fineness of the bow and the stern, the distribution of displacement was not intended to be that which would give the absolute minimum resistance for the various prismatic coefficients. It was admitted in the paper that in the fuller models the bow is too bluff at any except low speed. It is intended to give an idea as to the variation of power and propeller efficiency with fullness of lines. The general trend of these variations might differ a little if absolutely the best model were chosen for each displacement, but the variation as to the prismatic coefficient to use under different conditions would not be very much, and it is believed that the data given in the paper may be used with considerable confidence in determining which is a proper prismatic coefficient to use for a single-cargo ship under a given set of conditions.

With regard to propeller efficiency and to the statements by Mr. Robertson and Professor Sadler that a propulsive coefficient of 65 per cent appears high, I am perfectly willing to admit that I was rather surprised at the relatively high efficiency obtained when I first examined the data resulting from the tests. It must be remembered, however, that propulsive coefficient in the present paper refers to the ratio between the effective horse-power and the shaft horse-power and not to the ratio between the effective horse-power and indicated horse-power, as the term is most commonly used. Accurate trial data giving power and propeller revolutions and speed over a measured course under ideal conditions are very meager. However, attention is invited to last year's paper and to the discussion of the same by Rear Admiral C. W. Dyson. Admiral Dyson has collected data from the actual trials of ships for use by him in designing propellers. I asked Admiral Dyson to prepare some estimates for the ships represented by the models in last year's paper, which he kindly did and included in his discussion. The estimates prepared by him checked remarkably well with the results of the model tests. It must be remembered also in discussing propulsive efficiency that the results here given do not include shaft friction or loss in the thrust bearings.

That a propulsive efficiency as high as 65 per cent may be obtained in the ordinary cargo ship, with propeller running at about 90 revolutions per minute, seems to be confirmed by experiments along a somewhat similar line by Mr. Semple in England. A reference to his work is given in the body of my paper.

I was very much interested in the comments of Mr. Emmet, especially as to the variation of torque encountered at sea when a vessel is pitching. Though this variation undoubtedly must contribute to some cases of failure of reduction gears installed for ship propulsion, I am inclined to think that an equal number of failures may be ascribed to torsional vibration of the propeller shaft.

The President:—This is a practical paper. Taken in connection with the paper of yesterday by Mr. Robertson, it gives all those who operate ships food for thought. Although the paper is expressed in scientific terms, it has an exceedingly practical bearing. The comments of Mr. Robertson, Mr. Emmet, and Professor Sadler are supplementary and illuminating. We would like to hear something from any ship operating representative.

In the absence of further discussion, I am sure you will permit the Chair to present the thanks of the Society to Commander McEntee for his paper. It is most admirable in substance and form. It has had, besides, an illuminating discussion.

The next paper is entitled "Buoyancy and Stability of Troop Transports," by Professor William Hovgaard, Member.

Professor Hovgaard presented the paper.

BUOYANCY AND STABILITY OF TROOP TRANSPORTS.

By Professor William Hovgaard, Member.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

We shall here deal in particular with ocean-going transports such as those used to carry troops across the Atlantic during the world war. The number of troops on board a transport often greatly exceeds the number of passengers carried on a liner of the same size, and this fact, in connection with the urgency and vital character of military expeditions in general, makes the problem of the safety of troop transports rank higher in importance than in the case of ordinary passenger steamers.

REQUIREMENTS FOR SAFETY.

In order to obtain a basis for the discussion it is necessary to make certain assumptions as to the amount of damage which it may reasonably be stipulated that troop transports should be able to stand without their safety being imperilled. In addition to the ordinary dangers of navigation, notably collision and grounding, such vessels are in time of war exposed to attack from artillery, mines, and torpedoes, but we consider here those dangers only in so far as they affect the buoyancy and stability, involving damage below the bulkhead deck. In this connection underwater explosions are the most important.

Evidently the minimum claim to buoyancy and stability of a troop transport is that such a vessel should stand the effect of one underwater explosion or one collision without sinking or capsizing. But this is not sufficient. In order to provide for roughness of the sea and allow the safe lowering of the boats, there must remain a certain margin of reserve buoyancy and the ship must not take too great a list. Since an explosion or a collision is liable to damage one of the main transverse bulkheads, those requirements must be satisfied even if two adjoining compartments are flooded, whatever the conditions of loading. Finally, the spacing of the bulkheads should be such that there will be very little chance of two bulkheads being damaged by one explosion.

As explained hereafter, the claims stipulated above practically determine a minimum length of troop transports. It is at once clear, moreover, that in order to prevent a ship from taking an excessive list in the damaged condition, she must be subdivided essentially on the transverse system and the metacentric height must not fall below a certain minimum dependent on the size and design of the ship and the condition of loading.

BUOYANCY.

Under this head we discuss the spacing of the transverse main bulkheads with special regard to the danger of foundering by bodily sinkage, eventually combined with longitudinal inclinations, which may cause the ship to go down by the bow or the stern. We disregard the presence of longitudinal bulkheads and the transverse inclinations which they may produce.

1. *Mode of Procedure.*—In case of ships for which all the design data, as well as the conditions of loading, are accurately and completely known, the problem is relatively simple. The effects of bilging on draught and trim can be calculated by direct methods, all combinations of flooding of two or more compartments in accordance with the stipulated requirements being studied. There is, in fact, nothing in the problem which calls for an explanation beyond that already given in textbooks on naval architecture. The difficulties arise when it is required on short notice to equip and use, for transport service, ships such as ordinary transoceanic steamers about which only incomplete information is at hand and where circumstances do not allow time or opportunity for complete calculations. Such were the conditions in case of many passenger steamers taken over in 1917 by the United States Navy for use as troop transports, especially the German vessels, for which in general no line drawings, general arrangement plans, or weight statements were available. In all such cases it is necessary to resort to approximate methods of investigation.

Specific problems as to the effect of flooding of certain compartments may be solved directly by rough calculations, using approximate methods. In the Leviathan, for instance, it was found that flooding of all four engine-rooms together with adjacent side compartments would produce a bodily sinkage of about 4 feet and a total change of trim by the stern of 14½ feet. In the Mount Vernon a similar state of bilging would produce a sinkage of 3¾ feet and a total change of trim by the stern of 11¼ feet. This mode of investigation, however, cannot be applied with advantage to all possible cases of flooding of one or more compartments in a ship, because it would involve an amount of labor that would hardly be warranted in view of the crudeness here assumed of the available data. A more general and simple method is needed when it is required to deal with a great number of ships in a short time, in which case broad, comparative results are of more interest than absolute quantitative values.

2. *The Floodable-Length Method.*—We shall here briefly describe the so-called "floodable-length" method devised by the Committee on Subdivision of Merchant Ships, appointed by the British Board of Trade.* This method enables a designer with relatively small work to study the spacing of the bulkheads in a new ship with a view to the fulfillment of given requirements as to subdivision, and provides a

*The provisions of the International Convention for the Safety of Life at Sea relating to Safety of Construction of Ships are in substantial agreement with the results of the investigations of the Board of Trade Committee, which made its report in November, 1914.

ready means of determining the status in this respect of an existing ship, giving a measure of its safety as well as suggestions for eventual alterations in the watertight subdivision.

In principle the method consists in determining the "floodable length" at any point in a ship, being the length of hold, having its center at that point, which can be flooded without the ship foundering. The spacing of the bulkheads is expressed as a fraction of the floodable length obtained by multiplying it with a "factor of subdivision." If that factor is everywhere equal to or smaller than unity, but still greater than one-half, any one but not any two compartments may be flooded without the ship going down, and we have what is commonly called a "one-compartment" ship. If the factor is everywhere equal to or smaller than one-half but still greater than one-third, any two but not any three adjoining compartments may be flooded without the ship going down, and we have a "two-compartment" ship and so forth.

The floodable length is found as described in Vol. I of the Report of the Committee, while Vol. II gives a number of diagrams from which it may be obtained directly, expressed as a percentage of the length of the ship. The so-called "margin line" forms the basis of the calculation. It is drawn 3 inches below the deck-at-side line of the bulkhead deck, the latter being defined as the uppermost continuous deck to which all transverse watertight bulkheads are carried. The floodable length is so determined that, when that length of the hold is flooded, the ship shall not be submerged beyond the margin line.

Various elements are used as arguments in the diagrams of Vol. II for finding the floodable length, the most important being the freeboard ratio, which is the ratio between the freeboard to the margin line amidships and the draught amidships. Greater freeboard at the ends is taken into account by the sheer ratio, which is the ratio of the sheer of the margin line at the forward or after end to the draught amidships, the sheer being measured from the horizontal line through the lowest point of the margin line. The element next in importance is the permeability, which is the percentage of a given space, that can be occupied by water. When a ship is in the light condition, the high permeability of the empty hold spaces to some extent neutralizes the favorable effect of the greater freeboard, but nevertheless the floodable length is ordinarily much greater than in the full-load condition. Exceptionally the floodable length in the light condition may fall below that in the full-load condition at the ends. The numerical value of the permeability should be calculated very carefully, as it exerts a great influence on the result. The diagrams in Vol. II of the Report are prepared for permeabilities of 60 and 100 per cent. The floodable length for other permeabilities must be obtained by interpolation. The form of the hull is taken into account by using the block coefficient as one of the arguments in the diagrams, which are calculated for a standard form of various degrees of fineness. If ships differ materially from the standard form, certain corrections must be applied.

When the floodable length is obtained for a number of stations in a given ship or design, the results are marked up as ordinates from the base line of a profile drawing, which shows the location of the bulkheads, and a "floodable-length curve" is drawn through the points so obtained, giving together with the bulkheads a graphical representation of the safety of the ship in damaged condition.

In Plate 54 such curves are given for one of the ex-German ships used as a troop transport during the war. The lower curve corresponds to the full load, the upper to the light condition.

All floodable-length curves are characterized by a maximum amidships, a minimum on about one-quarter of the length from each end, and a pronounced rise towards the ends of the ship.

The floodable-length method, being applicable to all sizes and types of vessels, cannot, of course, be expected to give accurate results in any specific case. Nor is this claimed for it, but, used intelligently, it affords a valuable means of dealing in a practical and comparatively simple manner with problems relating to the spacing of bulkheads. It enables the naval architect to judge whether a given ship is suitable for use as a troop transport and to determine what alterations, if any, should be made in it to improve the safety. It was applied to all vessels used as trans-atlantic troop transports by the United States Navy during the war and showed that all the ex-German ships when in the light condition were two-compartment ships, many of them with an ample margin, and some approached or reached the three-compartment standard. In the full-load condition most of those vessels came up to the two-compartment standard.

3. *Changes in the Subdivision of Existing Ships.*—In cases of emergency ships may have to be used which are far from satisfactory in point of subdivision, but it must be borne in mind that other qualities have also to be considered. High speed and good maneuvering capability, which materially reduce the dangers of attack by submarines, may outweigh defects in subdivision. A well-subdivided but slow vessel may be more exposed to destruction by submarines than a poorly subdivided but faster vessel. Also the accommodations immediately available in a ship for carrying troops have to be considered.

If the subdivision of an existing ship is found unsatisfactory, conditions may be often materially improved by carrying one or more bulkheads up to the deck next above the bulkhead deck. This applies in particular to ships where non-watertight bulkheads, which can be readily made watertight, exist above the bulkhead deck as extension of some of the main bulkheads. The addition of one or a small number of new watertight bulkheads and slight modifications of bulkheads already existing above the bulkhead deck will be then sufficient to raise the bulkhead deck for the whole or part of the length of the ship, and the safety may be thus immensely increased. In some cases the subdivision below the bulkhead deck may with advantage be supplemented by additional bulkheads. Alterations of this nature were made by the Bureau of Construction and Repair in a number of transports, but on account of the urgency of the service and the short stay in port of the vessels no extensive changes could be carried out.

4. *Spacing of Bulkheads in New Designs.*—The claim stipulated above, that a troop transport should be able to stand the effect of one underwater explosion, determines the two-compartment standard as the minimum requirement, but where the size of a ship allows a higher standard to be reached, the designer should of course take advantage of this fact, even when that standard cannot be maintained throughout the entire length of the ship.

In determining the spacing, the designer must take into account the probable extent of the damage, the probability of bulkheads being lost, and the effects on the buoyancy. He must consider also the limitations imposed by the size and length of the ship, as well as by the general requirements of the service. We shall take up each of these questions separately.

(*a*) The first thing to consider is the horizontal extent of the zone of rupture and deformation caused by the explosion of a mine or torpedo, in so far as those effects destroy the integrity of the bulkhead. Its value varies greatly with the size of the charge and with various other conditions, but seems to lie between 25 and 55 feet. We shall denote this quantity by e and assume that under present conditions its probable highest average value is 35 feet. This is but slightly smaller than the figure adopted by Mr. W. S. Abell, Chief Ship Surveyor of Lloyd's Register of Shipping, in a paper read before the Royal Society, January 22, 1919, discussing the same subject for cargo vessels.

(*b*) There is for every part of a ship a lower limit to the spacing of the bulkheads, determined with regard to proper housing and working of the machinery, the service of the ship, the accommodation of the troops, and the stowage and handling of cargo as well as the proper utilization of space in general. In the forward and after holds, excepting the extreme end compartments, the spacing seldom falls below 35 feet in practice, and probably 40 feet may be considered as the smallest desirable spacing of the bulkheads in those parts of the ship. In the midship portion, where the machinery is located, it is often necessary to make the compartments much larger, especially in ships like troop transports which have or should have great engine power.

(*c*) If the spacing of the bulkheads is equal to e, the extent of the damage caused by an explosion, it is likely that an attack would always injure at least one bulkhead, but by increasing the spacing beyond this limit even by a very moderate amount, the probability of a bulkhead being lost is very much reduced. Let the spacing be s, then this probability, which we shall call c, is measured by the ratio between the danger space, e, and the remaining space, $s - e$, in each compartment. Hence—

$$c = \frac{e}{s-e} \qquad (1)$$

When $s = e$, we have $c = \infty$, but when s is 20 per cent greater than e, $c = 5$, that is, the chances are only 5 to 1 that a bulkhead will be lost by a single explosion.

Based on these considerations it is proposed in troop-transports to adopt a minimum spacing of the bulkheads $s_0 = 45$ feet, which gives a maximum value of the probability factor:—

$$c_0 = \frac{35}{45-35} = 3.5.$$

(d) In order to secure the two-compartment standard, the combined length of any two adjacent compartments must in no case exceed the floodable length, f, and should in general be slightly smaller. Hence—

$$2s \leqq f \tag{2}$$

and the smallest permissible floodable length will be—

$$f_0 \geqq 2s_0, \text{ or}$$

$$f_0 \leqq 90 \text{ feet} \tag{3}$$

When the floodable length at any point is so great as to admit of a subdivision into three compartments, each of which has a length greater than s_0, the three-compartment standard, should be adopted in that region if practicable. In such a case no single explosion is likely to flood a greater length than $\frac{2}{3}f$, and a margin of safety will remain, which enables the ship to sustain further damage without sinking. A similar procedure may be followed where practicable in cases where $f > 4s_0$, resulting in a still greater margin of safety.

(e) If we take into account not only the direct effects of flooding but the probability of a bulkhead being lost, there is a limit to the closeness of spacing of bulkheads, which it is not profitable to pass. In other words—contrary to what might appear at first sight—it is not always advantageous from the point of view of safety to space the bulkheads closer together.

Consider first a one-compartment ship, where the loss of a bulkhead means the loss of the ship. If the spacing is here made equal to e, there is a practical certainty that the ship will go down if she is hit by a torpedo. If the spacing is 45 feet, that is $s = s_0$, we have seen that the probability of a bulkhead being lost is as 3.5 to 1. If the floodable length is 90 feet, the limiting value for the one-compartment standard, then by increasing s beyond the 45 feet, the probability of a bulkhead and hence of the ship being lost is reduced and reaches its minimum when $s = 90$ feet:—

$$c = \frac{35}{90-35} = 0.64.$$

In general, the greater the spacing, the smaller the probability of loss, and the maximum safety within the one-compartment standard is evidently attained by making the spacing equal to or slightly smaller than the floodable length.

Suppose next that a ship is designed to the two-compartment standard where the loss of one bulkhead does not endanger the safety of the ship. We have then, with a minimum spacing of s_0:—

$$2s_0 \leqq f \leqq 3s_0$$

or, with the value of s_0 here adopted—

$$90 \text{ feet} \leqq f \leqq 135 \text{ feet}.$$

With a spacing of $s = s_0 = 45$ feet, we have again the probability of a bulkhead being lost: $c = 3.5$, but gradually, as s is increased, c will fall off, until in the limit when f is equal to 135 feet and s equal to say 65 feet, we have

$$c = \frac{35}{65-35} = 1.17,$$

that is, the chance of a bulkhead being lost is one-third of what it would be with $s = 45$ feet. This advantage of the wider spacing is believed to outweigh the drawback that when a bulkhead is lost and two compartments flooded there will be a smaller margin of safety than with the shorter spacing. Hence it is best in a two-compartment ship always to make the spacing approach one-half of the floodable length.

When $f > 3s_0$ or $f > 135$ feet, the three-compartment standard can be attained. Suppose that—

$$135 \text{ feet} < f < 180 \text{ feet},$$

then the best result is attained within the three-compartment standard by making s equal to or closely approaching $\frac{f}{3}$, in which case the probability of a bulkhead being lost will range from 3.5 to 1.4. It might be argued that it would be better to retain the two-compartment standard and make $s = \frac{f}{2}$, since then the value of c would be much reduced and would range from 1.08 to 0.64, but in that case the hold compartments would be unnecessarily, perhaps inconveniently large, and when a bulkhead was lost the flooded length would be 50 per cent greater than with the three-compartment ship. It is to be borne in mind also that a three-compartment ship is better able to stand underwater damage caused by collision or gunfire which is not so liable to destroy the bulkheads. Hence, when f is greater than $3s_0$ and smaller than $4s_0$, the three-compartment standard should be adopted with s equal to or slightly smaller than $\frac{f}{3}$.

It is clear that the same reasoning will apply and an analogous conclusion will be reached when the floodable length is greater than $4s_0$ and we arrive at the following general rules:—

1. The standard of subdivision should be the highest consistent with the floodable length and with the adopted minimum spacing of the bulkheads. If $ns_0 \leqq f \leqq (n + 1)\, s_0$, the n-compartment standard should be adopted.

2. Within the given standard of subdivision, the spacing of the bulkheads should be the greatest possible consistent with that standard, provided no practical considerations stand in the way of such spacing. With the n-compartment standard the spacing should closely approach $\frac{f}{n}$. In other words, the factor of subdivision according to this rule is practically either unity or some simple fraction:—½, ⅓, ¼, etc.* It should be observed that the same ship may be, at the same time, subdivided to different standards, depending on the value of the floodable length at various points. Frequently ships, which may be subdivided to the three-compartment standard at amidships and at the extreme ends, can be subdivided only to the two-compartment standard on the quarter length, where the floodable length is a minimum, and such vessels will be referred to as two-compartment ships.

(f) In troop transports, which are liable to be loaded more deeply than ordinary passenger vessels, the floodable lengths are relatively small. In the following table are given the average minimum values of f as found in troop transports of good design used during the war by the United States Navy. The minima occurred as usual on about one-quarter of the length of the ships from the ends. In the third column are given what are considered appropriate values for the corresponding spacing of the bulkheads at the points when the floodable length is a minimum.

TABLE I.

Length of ship, L	Minimum floodable length, f	Spacing of bulkheads, s	Standard of subdivision
Feet	*Feet*	*Feet*	
400	70	65	One-compartment
500	90	45	Two-compartment
600	110	50—55	Two-compartment
700	125	60	Two-compartment
900	160	50	Three-compartment

Ships of less than 500 feet in length cannot, according to this table, be subdivided in a satisfactory manner to the two-compartment standard throughout their length and are not, therefore, suitable for the transport of troops. Ships of more than 500 feet in length may be subdivided according to the two-compartment stand-

*This rule differs from the recommendations of the Board of Trade Committee on Subdivision of Merchant Ships in that the factor of subdivision, according to the committee, bears no definite relation to the floodable length, being obtained from a continuous curve plotted on the length of the ship as abscissae.

ard and may reach the three-compartment standard amidships, but the latter standard can probably not be attained throughout and to full advantage in ships of less than 800 or 900 feet in length. Higher standards relative to the length of the ship may be attained, but only by carrying the bulkheads up to a rather unusual height or by a reduction in the carrying capacity of the ship, that is, by a disproportionate height of freeboard of the bulkhead deck and a low draught. One-compartment ships may of course be employed for the transport of troops in an emergency, but should ordinarily be used only as supply vessels or other auxiliary duty.

Differences of opinion will no doubt exist as to the numerical values to be adopted for the probable extent of damage by an explosion, the practicable minimum spacing of bulkheads, and the assumptions on which the floodable lengths should be calculated, but this does not affect the validity of the principles here stated. The main results of the investigation, as relating to troop transports for ocean service and on the present standpoint of submarine weapons, may be summarized as follows:—

1. Troop transports should have a length of not less than about 500 feet.

2. The freeboard of the bulkhead deck and the sheer should be such that, when the ship is on her deepest draught and with reasonable assumptions as to permeability, she will conform at least to the two-compartment standard throughout her length.

3. Excepting the extreme ends of the ship, the spacing of transverse main bulkheads should not in general be less than about 45 feet.

4. Whichever standard of subdivision is adopted, the spacing of the bulkheads, at any point of the ship, should be the greatest practicable consistent with that standard.

5. *Steps in bulkheads.*—When a transverse bulkhead is divided into parts placed on different frames, it is said to be stepped. Such a bulkhead actually consists of two or more vertical parts and one or more horizontal parts, or steps, the latter being formed by the deck or decks at which the continuity of the bulkhead is broken. Damage to the side by explosion or collision in way of a step located below the waterline is likely to open both of the adjacent compartments to the sea and thus, dependent on the length of the step, the value of the subdivision is seriously impaired. Suppose, for instance, that, as indicated in Fig. 1, a bulkhead has a step 35 feet long and that the horizontal extent of the damage by an underwater explosion is likewise 35 feet; then the chance of both adjacent compartments being flooded by one explosion is practically doubled due to the presence of the step.

Steps in bulkheads are therefore to be avoided or their lengths at least reduced to a minimum.

Recesses or horizontal steps in bulkheads are less objectionable, provided they do not extend so near to the sides of the ship as to be damaged by an underwater explosion.

6. *Integrity of Bulkheads and Other Watertight Surfaces.*—Several precautions must be taken to ensure the effectiveness of watertight subdivisions, having for

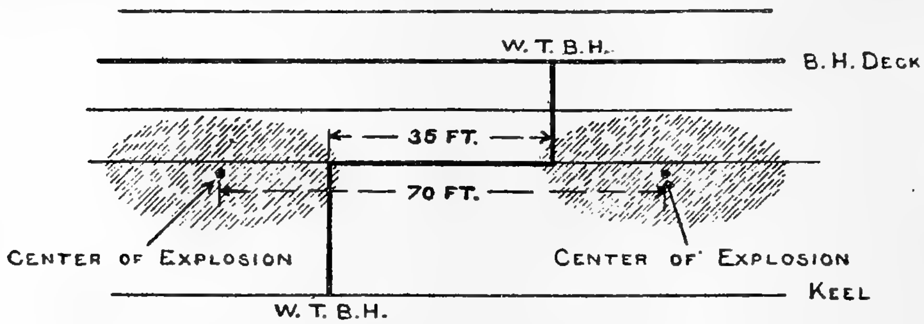


FIG. 1.—STEPPED BULKHEAD.

their main object to prevent the water from entering compartments other than those directly and unavoidably flooded by an accident.

In merchant vessels steps in bulkheads involve a danger in addition to that just explained, due to the fact that the frames almost invariably pass non-watertight through all decks below the bulkhead deck. When one of the compartments adjacent to a stepped bulkhead is flooded, and provided the step is situated below the water-line, water is liable to filter through the deck at the frames. The cement filling usually applied along the sides of the deck cannot be relied upon to keep tight under a considerable head of water. Thus the integrity of the stepped bulkhead is destroyed. The remedy is to fit stapling around the frames in that part of the deck which belongs to the bulkhead.

Recesses in bulkheads are not objectionable from this point of view as they are usually watertight at the bounding decks.

The danger of fitting doors in the main transverse bulkheads of troop transports is enhanced by the difficulty of controlling the doors at all times. This may be due partly to the overcrowded condition liable to occur in troop transports, partly to the impossibility of obtaining a highly trained personnel for all such vessels under the stress of war. Great care should, therefore, be bestowed on this matter, and no doors should be allowed in main transverse bulkheads below the bulkhead deck, but this rule is not followed in passenger steamers. Doors are found partly on a low level as between engine and boiler-rooms, partly on a high level, notably on the deck just below the bulkhead deck. When such vessels are taken over for service as troop transports all doors in transverse bulkheads on a low level in the engine and boiler-rooms should be permanently closed. This, of course, renders communication and inspection more difficult and many engineers for this reason strongly object to it. It is, however, a fact that, in some of the largest American transports used during the war, those doors were always closed when at sea without serious detriment to the service.

In most transoceanic passenger steamers the hold just forward of the boiler-rooms is used as a reserve coal bunker. Often two compartments of the fore hold are so used. Coal is drawn from the reserve bunkers through doors in the respective main bulkheads at the level of the fire-room floor, and often a tunnel with a door at each end is fitted through the main transverse bunker usually fitted in the forward end of the forward boiler-room, so that coal may be drawn directly from the nearest reserve bunker. Such doors, putting two of the largest compartments in the ship in connection with each other and being necessarily open during long periods, while the ship is at sea, constitute a serious menace and should be closed under war conditions. This necessitates hoisting the coal from the reserve bunkers on to the bulkhead deck, whence it can be dumped into one of the regular bunkers, an arrangement which was adopted in several ships during the war.

The transverse main bunker bulkheads in passenger steamers are generally non-watertight with non-watertight sliding bunker doors and should be disregarded in considering the watertight subdivision, but even if such bulkheads are of watertight construction, it is best not to rely on them, since the doors will normally be open when the ship is under steam.

The following further recommendations are essentially an abstract of those given for large mail and passenger vessels under war conditions by a committee appointed by the Institution of Naval Architects in 1917 to inquire into the effects of underwater explosions on the structure of merchant ships.*

Firemen's passages should be permanently closed up and all openings from the engine-rooms into shaft tunnels should be abolished, access to tunnels being provided through watertight trunks abaft the engine-room bulkhead. If these recommendations are not carried out, the engine-room tunnel doors should be quick-closing and capable of operation from the bulkhead deck or a deck above.

Doors fitted in main bulkheads on the 'tween deck below the bulkhead deck cannot always be dispensed with without great inconvenience to the service and without radical alterations. It is a fact, however, that such doors are very dangerous in cases where a ship takes a sudden and great list, whereby water is admitted to the 'tween deck and tends to spread forward and aft (Empress of Ireland). It is advisable, therefore, to close those doors, to secure them so that they cannot be opened, and to provide additional exits to the decks above where necessary.

The committee further recommended that all side scuttles situated below the first deck above the bulkhead deck be closed up and sealed. It may here be added that all side lights below the bulkhead deck should be not only closed but also protected by steel plates or strong watertight covers, as they are otherwise liable to be broken by the force of the underwater explosions.

Valves should be fitted on all sanitary discharges at the ship's side such as will prevent water passing inboard through them when the vessel has a considerable trim or list.

The upper ends of ash or rubbish shoots, etc., opening on decks below the bulkhead deck should be provided with watertight covers, which should always be in place when the shoots are not in use.

*Report of May 8, 1917. Institution Naval Architects, 1918, p. xxxvii.

Each suction pipe, where it enters the compartment from which it drains, should be provided with a screw-down, non-return or other suitable valve, which can be worked from the bulkhead deck. These valves should be kept closed except when the pumps are in use.

Where ventilation trunks, etc., pass through watertight bulkheads below the bulkhead deck, valves operated from above that deck should, in all cases, be fitted at the bulkheads. No such trunks should be allowed to pierce the bulkheads at a low level.

The precautions here enumerated were carried out as far as practicable in the transports operated by the United States Navy in accordance with general orders to that effect issued by the Bureau of Construction and Repair.

STABILITY.

The stability of a troop transport should be such that under the above stipulated conditions of flooding there will remain a certain positive metacentric height, while the list, if any, must not be greater than that the boats can be lowered on both sides. In order to ensure this result an investigation of the stability must be made. This presents no difficulty where information is at hand concerning the lines and the distribution of weights and where an inclining experiment has been made, but where such information is not available and where time is limited, approximate or crude methods must be adopted. We shall here describe the procedure followed by the Bureau of Construction and Repair during the war in such cases, as occurred quite frequently when merchant vessels in an emergency were taken over by the Government and used as troop transports.

1. *Combined Inclining and Rolling Experiment.*—The first point to consider was the metacentric height under various conditions of loading when the ship is intact. A fair idea of the form of the hull was obtained by lifting offsets of the bottom while the ship was in dock or, when this could not be done, by taking internal measurements. When practicable, an inclining experiment was made, and at the same time the period of roll was determined. A combined inclining and rolling experiment gives the value of the coefficient K in the formula:—

$$T = \frac{KB}{\sqrt{GM}}$$

Where T is the time of a single swing in seconds, B is the beam and GM the metacentric height, both in feet.

This formula affords a ready means of estimating the metacentric height at any time even when at sea, by simply observing the period of roll.

Table II gives the results of such experiments for a few ships.

TABLE II.—*Results of Combined Inclining and Rolling Experiments.*

Ship	Displacement at time of experiment T	GM	T	K	D
	Tons	Feet	Seconds.		Feet Inches
America ex. Amerika....................	27,500	1.4	14.0	0.220	27 6
Mount Vernon ex. Kronprinzessin Cecilie..	27,250	2.5	10.2	0.223	30 9
Covington ex. Cincinnati................	22,000	2.5	9.15	0.221	28 9

The America, during the experiment, was near her light condition, although she had 4,300 tons of coal and 1,450 tons of water on board. The Mount Vernon was on about the same draught as she would be carrying troops across the Atlantic, when on her eastbound voyage and nearing a European port; she had 4,650 tons of coal and 2,570 tons of water on board. The Covington was fully equipped, but, like the two other ships, without troops or cargo.

2. *Various Conditions of Loading.*—A careful investigation was made of the probable loading of the ships under various conditions, and the corresponding draught and trim were calculated. In particular, the following conditions were studied:—

(1) When a transport sails from the port of embarkation, fully equipped, with troops and cargo on board.

(2) When she arrives at the port of debarkation.

(3) When she sails from that port.

(4) When she returns to the port of embarkation.

For each of these conditions the position of the center of gravity and the metacentric height were calculated, or, where no inclining experiment was available, the position of the center of gravity was determined by an approximate calculation.

3. *The Period of Roll at Sea.*—In all cases, whether an inclining experiment was made or not, the period of roll was observed later when at sea in the various conditions of loading. As far as possible the observations were taken under circumstances when the rolling was of a fairly regular character and small or moderate in amplitude. By comparison with other ships of similar size and type for which K was known, this coefficient was estimated for each vessel and the metacentric height calculated approximately from the formula:—

$$GM = \frac{K^2B^2}{T^2}.$$

It is also possible in a general way to judge of the stability of a ship directly from the observed period by comparing it with that of other similar ships. This method was actually used in the transports under the control of the United States Navy during the war and afforded a ready and valuable means of controlling the loading and ballasting. In fact, observation of the period at sea was the only available method by which it was possible to obtain a knowledge of the stability in the majority of cases. The following table gives the observed period of roll and the mean draught for a number of typical ships in full-load and light conditions, that is, at the beginning and end of the round trip.

TABLE III.—*Periods of Roll at Sea.*

Ship	After leaving U. S. Full load			Before arriving U. S. Light		
	D		*T*	*D*		*T*
	Feet	Inches	Seconds	Feet	Inches	Seconds
Leviathan ex. Vaterland	40	11	11.0	36	8	13.0
George Washington	35	0	8.0	28	5	11.0
Mount Vernon ex. Kronprinzessin Cecilie	34	6	9.0	26	0	11.0
President Lincoln	32	7	8.7	24	3	6.7*
Covington ex. Cincinnati	30	9	8.1	24	6	9.9
Louisville ex. St. Louis	30	4	7.7	27	9	8.2
Æolus ex. Grosser Kurfürst	26	11	6.5	22	9	9.0
Pocahontas ex. Prinzessin Irene	26	9	6.0	23	3	7.5
Great Northern	26	8	7.4	21	8	7.8
Calamares	24	4	9.0	19	6	11.3
Orizaba	24	0	7.0	18	3	11.0

4. *The Loss of Stability by Bilging and the Residual Metacentric Height.*—The effects of bilging on stability may be studied by calculating the change in metacentric height and the angle of heel by approximate methods. In ships subdivided on the purely transverse system it is sufficient to determine the change in metacentric height, assuming that two or three of the largest compartments are flooded, according as the ship is required to come up to the two or three-compartment standard respectively. In order to obtain the severest case, it may be necessary to repeat the calculation for several combinations of flooded compartments. Using the lost buoyancy method, by which the center of gravity remains fixed in position, the change in metacentric height, MM_1, is the algebraic sum of the vertical movement of the center of buoyancy, BB_1, which is always positive, and the change in metacentric radius, which is always negative, and approximately equal to $\frac{si}{V}$, where i

*President Lincoln, in the light condition, carried 2,400 tons of steel ballast, 2,200 tons of water, and 1,700 tons of coal.

is the moment of inertia of the free surface of water in the flooded compartment, s is a factor of permeability, and V is the volume of displacement of the ship. We have, therefore, in a transversely subdivided ship—

$$MM_1 = BB_1 - \frac{si}{V}.$$

When a ship is in the light condition, BB_1 is relatively small, because the water does not rise to a great height inside the ship, but $\frac{si}{V}$, which represents the effect of the free surface of water in the compartment, will have approximately its full value. The loss in metacentric height is, therefore, a maximum and, hence, the ship is in this condition most liable to take a great list. In the full-load condition, on the other hand, BB_1 is very great, while $\frac{si}{V}$ has about the same value as in the light condition, so that MM_1 is either negative and relatively small or, in some cases, even positive. Hence, after flooding, the metacentric height in the full-load condition is practically the same as when the ship is intact and, provided there are no longitudinal bulkheads, the ship will remain upright with ample stiffness; but, as shown above, in this condition the danger of foundering by bodily sinkage or of going down by the bow or stern is at its maximum.

Calculations made for a great number of transports showed that the loss in metacentric height by flooding two of the largest compartments in two-compartment ships when in the light condition varied from 1 to 2 feet, being ordinarily about 1½ feet. In most vessels flooding of the forward boiler-room, together with the adjoining forehold—generally used as a reserve bunker—gave the greatest loss in stability.

Having calculated the probable loss in metacentric height in a given ship, it remains to estimate what the residual metacentric height ought to be after bilging, whereupon the appropriate value of this element in the intact condition can be determined. Evidently the residual metacentric height must be estimated with due regard to the size of the ship and her liability to take a list.

Ships that are subdivided on the purely transverse system will remain upright so long as there is any stability left and no external forces tend to heel them over. They will be safe, therefore, with a very small metacentric height, say from ½ foot in vessels of about 500 feet in length to about 1 foot in vessels of the largest size. Assuming a loss by bilging of 1½ feet, the initial metacentric height should therefore be not less than from 2 to 2½ feet in the light condition while the ship is yet intact: If the calculated loss is greater than here estimated, these figures must be correspondingly increased. This requirement being satisfied in the light condition, there will generally be ample stability in the full-load condition.

Ships with longitudinal bulkheads, whether watertight or not, should have a greater residual metacentric height in damaged condition than required for ships with only transverse bulkheads. The residual metacentric height should be so calculated that under the stipulated conditions of damage the list shall not prevent the lowering of the boats on both sides of the ship, that is, the list should probably not be greater than from 5 to 10 degrees. It must be borne in mind, however, that when holes are provided in the longitudinal bulkheads, as recommended above, a rapid equalization will take place immediately after the accident and the list may be expected to decrease very fast during the first few minutes, unless other causes, tending to increase it, intervene. It seems reasonable, therefore, in estimating the residual metacentric height, to allow a somewhat greater angle of heel, depending on the nature of the longitudinal subdivision. Assuming a permissible initial list of from 10 to 15 degrees, it is found that the residual metacentric height should probably be not less than from 1½ feet in transports of about 500 feet in length to 2½ feet in transports of the largest size. This gives a metacentric height in the light intact condition of from 3 to 4 feet, but these figures must be regarded only as average values, applicable to ships of existing type suitable for that service.

In the full-load condition the freeboard and hence the range of stability is a minimum, but, on the other hand, the metacentric height is likely to be greatest and does not in general suffer so much reduction by bilging. Again, if the stability is sufficient in the light condition, it will generally be sufficient when the ship is fully loaded, but probably the metacentric height should not be less than from about 4½ feet in smaller transports to 5½ feet in transports of the largest size in the intact full-load condition. Here, as when in the light condition, each ship must be dealt with independently and on its own merits.

As might be expected, these requirements are not fulfilled in ordinary passenger steamers as operated under peace conditions. The Empress of Ireland, which, as described below, was lost in a collision, had a metacentric height of 3½ feet when the collision occurred, at which time she was practically in the full-load condition. This was reduced to 2¾ feet* by flooding of the boiler rooms. In a troop transport of the same size the metacentric height should be about 1 foot greater. Had doors and side lights been closed on the main deck and had holes been provided in the side-bunker bulkheads, the list would soon have been reduced by equalization to less than 10 degrees, in which case the ship would probably have remained afloat and boats could have been lowered on both sides.

The George Washington, which has no longitudinal subdivision, satisfies practically the above requirements, having a metacentric height of 2½ feet in the light condition and 4½ feet in the full-load condition, but in the Leviathan these figures are 3 feet and 4 feet and in the Mount Vernon 2.1 feet and 3.2 feet respectively, showing that these vessels, which have many longitudinal bulkheads, fell short of the requirements.

BALLASTING.

In order to secure the metacentric heights specified above, most passenger steamers used as troop transports must carry some ballast, at least when they are in the light condition. Ocean-going passenger steamers are in general designed

*According to the lost buoyancy method.

with rather small stiffness. According to the best practice they are in the lightest condition given almost zero stability, that is, a very small positive metacentric height, in which case they may be expected to have a metacentric height in the loaded condition sufficient for safety under peace conditions and yet so small as to give easy movements in a seaway.* German designers of such vessels have gone further in direction of reducing the stability,† and most of the ex-German ships used as troop transports during the war have a negative metacentric height in the light condition unless they are given a considerable amount of ballast.

Ballast may be solid or liquid. Solid ballast usually consists in pig-iron, scrap-iron, cement, gravel, or sand and, if permanently on board the ship, is referred to as "fixed" ballast. Liquid ballast ordinarily consists of water, fresh or salt. The quantity of ballast should be determined by a careful calculation so as to secure the desired metacentric height under various conditions of loading, but when a ship is taken over in an emergency for immediate use, there is no time for elaborate calculations and the amount of ballast required must be estimated preliminarily, based on the behavior of the ship in commercial service and on a comparison with other ships. Later, when experience has been gained at sea, the ballast may be adjusted more accurately, and here the period of roll affords a valuable guide. This procedure was followed in most of the ex-German ships used as troop transports.

On the whole, water ballast is preferable to solid ballast, since it can be more readily taken on board and again discharged when not needed, and does not in general occupy useful space in the hold. Solid ballast should not be used except when the tanks available for water ballast are of insufficient capacity for this purpose. In many ships, however, the amount of ballast required in the light condition is much greater than it is possible or practicable to carry in the tanks, and it becomes necessary, in order to avoid shipping or unshipping of ballast on each voyage, to carry permanently a considerable weight of solid ballast, which in the full-load condition is a useless deadweight. One reason why tanks of sufficient capacity may not be available for the purpose of ballasting, even where the tank space is very large, is that when the troops are on board most of the tanks are used for fresh water and it may not be possible at the port of debarkation to obtain fresh water in sufficient quantity or of satisfactory quality. There is then no choice but either to leave the tanks empty on the return voyage or to fill them with salt water, but the latter course is clearly objectionable, as it necessitates washing out of the tanks at the port of debarkation, for which operation there may be no time or opportunity. Hence it is often preferred to leave most of the tanks empty on the return voyage and to carry fixed ballast instead.

The need of fixed ballast would, of course, disappear if the ships were designed so as to have sufficient stiffness under all conditions, and this solution is to be preferred in vessels designed and built exclusively for troop transport, where safety under war conditions is the first consideration. In passenger steamers, designed for

*L. Peskett, Institute of Naval Architects, 1914, p. 191.

†P. Driessen, *Schiffbau,* 1909, p. 269.

service in peace time, comfort is of prime importance, the stability must necessarily be small, and ballasting under certain conditions may become a necessity.

The following table gives the weight of fixed ballast carried by some of the troop transports during the war, and also the amount of water on board the ships when they were in the light condition near the end of their voyage.

TABLE IV.—*Ballasting of Troop Transports.*

Ship	Fixed ballast	Water, fresh and salt
	Tons	*Tons*
Leviathan	820	5020
George Washington	921	2290
President Lincoln	2400	2200
Æolus	1300	1670
Pocahontas	1300	1370

LONGITUDINAL BULKHEADS.

Most merchant vessels are subdivided on the purely transverse system. When damaged under water they usually preserve their upright position or take a slight list, and if they founder they go down by bodily sinkage, bow or stern first. The boats can be lowered without difficulty, provided the sea is not too rough, and in well-subdivided ships ample time will be generally left for this operation (Titanic). At the International Conference on Safety of Life at Sea, in London, 1913,* "evidence was produced which made it perfectly clear that there was no known case of a vessel having been bilged in service and then capsizing. * * *"

Longitudinal watertight bunker bulkheads commenced to be introduced in merchant vessels with the Lusitania—primarily in order to secure coal protection against gunfire—and were subsequently fitted in certain other English liners destined to be used as auxiliary cruisers in time of war. In Germany such bulkheads were fitted in several vessels as in the Kronprinzessin Cecilie, her sister-ship Kaiser Wilhelm II and the Vaterland, later used during the war as troop transports by the United States Government and re-named the Mount Vernon, Agamemnon, and Leviathan respectively. In these vessels center-line bulkheads, moreover, were fitted in the engine-rooms.

In other passenger steamers as, for instance, the S. S. Philadelphia, used as a troop transport under the name of Harrisburg, side bunkers are fitted which are virtually independent of each other, since they can only equalize across the top of the boiler-room on a level well above the deep-load water-line.

*Sir A. Denny, Institute Naval Architects, 1914, p. 204.

Side-bunker bulkheads, whether watertight or not, always constitute a danger to stability in merchant vessels where adequate means of quickly correcting a list are not available. An underwater explosion on one side may cause a bunker on that side to be flooded immediately and, generally, also the adjoining boiler-room. The opposite bunker will not as a rule be flooded so quickly, even if the bunker-doors are non-watertight and open, because usually the doors are choked with coal and the water can enter but slowly. The ship, therefore, at once takes a list, and although a gradual equalization may take place, water is likely at the same time to filter into adjacent compartments or spread over 'tween decks so that the stability may be destroyed and the ship capsize before equalization in the bunkers can prevent it. In such a case, then, the loss of the ship is directly due, not to the longitudinal bulkhead on the damaged side, but to the intact bulkhead on the other side, and the danger is greater the more watertight that bulkhead is. In point of safety side bunkers are useful only provided they are narrow, well subdivided, and watertight, and then only in cases of minor damage, where the side bulkhead remains intact. It is best, therefore, to avoid all such bulkheads in merchant vessels.

The danger of side bunkers in passenger vessels became evident by the loss of the Lusitania on May 7, 1915. The Lusitania was going at about 18 knots speed when she was hit almost simultaneously by two torpedoes on the starboard side abreast of the boiler-rooms. One of the torpedoes hit near the bulkhead separating boiler-rooms 1 and 2. Lifeboat No. 5, which is about 160 feet aft of that bulkhead, was destroyed by the blast of an explosion, but whether from the same or the other torpedo is unknown. The ship at once took a list of about 15 degrees, undoubtedly due to the presence of the side bunkers, and this list gradually and rapidly increased, due to infiltration of the water, especially through open sidelights. At the same time the bow was sinking deeper into the water than the stern. In less than twenty minutes after the explosion the ship capsized and sank. The initial list was so great that it was practically impossible to lower the boats on the port side, although the sea was calm. Due to this fact and the short time available before the ship went down, about 1,200 lives were lost out of a total of about 1,950 persons on board the ship.

Center-line bulkheads are no less objectionable and dangerous in merchant vessels than in warships. A calculation shows that, in the Leviathan, flooding of the two engine-rooms on one side will produce a list of 19 degrees, and in the Mount Vernon of 21 degrees, while in the George Washington, where there is no center-line bulkhead, flooding of the main engine-room will leave the ship upright, the sole effect being a sinkage of about 2½ feet with a total change of trim by the stern of about 5 feet.

A more insidious source of danger is found in many merchant steamers—even such as are ostensibly subdivided on the transverse system—due to the existence of "shifting bulkheads" fitted longitudinally inside transverse bunkers, usually at or near the center line and often on top of a tunnel. These bulkheads, which are fitted for the purpose of preventing the coal from shifting, are non-watertight and are in some ships provided with holes, which in case of bilging permit an equalization of the water from one side to the other. Ordinarily, however, the water must rise to a considerable height on one side before it begins to equalize, especially where the bulkhead is fitted over a tunnel and the holes are at a high level. Often the flow is very sluggish, because the openings are choked by coal. Thus shifting bulkheads, temporarily at least, have the same effect as if they were watertight.

A striking proof of the dangers of such quasi-longitudinal subdivision is furnished by the loss of S. S. Empress of Ireland in May, 1914, as the result of a collision with S. S. Storstad. The Empress of Ireland had two large boiler-rooms, each with a cross bunker at either end connected by side bunkers. All the bunker bulkheads were non-watertight. Water could flow freely from the side bunkers into the adjacent cross bunkers, but at the center line in these latter were fitted longitudinal passages and shifting bulkheads which, although provided with holes, greatly retarded the equalization of water from one side to the other inside the bunkers (see Plate 55). The ship was otherwise subdivided on the transverse system. At the time of the collision the bunkers were practically full of coal.

The Storstad struck the Empress of Ireland almost at right angles on the starboard side in way of bulkhead No. 5, which separates the two boiler-rooms from each other. A breach was produced in the side of the Empress of Ireland of an area below the water-line, estimated to be not less than 350 square feet—about the same size of hole as would be produced by the explosion of a torpedo. The immediate effect of the damage was that the watertightness of bulkhead No. 5 was destroyed and that both boiler-rooms were put in communication with the sea. Flooding of these compartments involved a mean sinkage of practically 9 feet, which took the main deck 4 feet below the water at the center line amidships. At the same time the ship took a heavy list.

Quoting from the Parliamentary Report, p. 17:—"There is no evidence that the Storstad destroyed any portion of the bunker bulkheads, so that very shortly after the impact a large quantity of water must have entered the bunkers on the starboard side for the whole length of the boiler-rooms, which water was able to escape only through bunker doors into the boiler-rooms and "relatively"* slowly also across the middle-line partitions in coal bunkers to the port side of the vessel. Under these circumstances the ship would at once commence to list to starboard * * * making reasonable approximations, an inclination of some 15 to 20 degrees appears probable. * * * From such a list the vessel might have recovered as the water got to the port side, if all port holes, and all watertight doors in bulkheads bounding the boiler compartments up to the upper deck, had been closed, but with doors and side lights open to the extent known to have obtained after the collision, water was free to enter other compartments, and the final capsizing and foundering became inevitable."

It is clear from this that it was owing to the presence of side bunkers and shift-

*Quotation marks are by the author.

ing bulkheads that the ship at once listed to such an angle that it was impossible to lower the boats on the port side and that equalization in the bunkers was delayed. Open side lights and bulkhead doors between the main and upper decks allowed the water to enter and spread forward and aft on the starboard side of the main deck, and before equalization could become effective the stability was wholly destroyed. About fifteen minutes after the collision the ship capsized and foundered with a loss of more than a thousand lives. This disaster happened in a perfectly calm sea and in a ship which was generally considered well designed and safe.

Longitudinal shifting bulkheads are fitted also in the cargo holds of all merchant vessels which carry cargoes that are liable to shift, but are in such case ordinarily provided with large door openings at the foot, and under the cargo hatches portable wooden partitions take their place, allowing a fairly free equalization of the water in case of bilging.

The obvious remedy against the ill effects of longitudinal subdivision is in all cases the same—to destroy the integrity of the longitudinal bulkheads by cutting permanent openings of ample size in them at the lowest possible level. In the center-line bulkheads of engine-rooms it will generally be sufficient to remove existing doors or to secure the doors in the open position so that they shall never be closed while the ship is at sea. If, however, the doors are of small area or placed at a high level, permanent openings should be cut at the foot of the bulkheads. Prof. C. Pagel, Director of the Germanischer Lloyd Classification Society, in 1916,* speaking of the danger of center-line bulkheads, made the following statement:—"Hence, for those of our (the German) passenger steamers that are provided with a center-line bulkhead in the engine-room, it is prescribed that in such a bulkhead great doors shall be fitted, which must be open during the voyage, so that, in case of bilging, water can flow to the endangered side and thus a great list be avoided." In the shifting bulkheads of transverse bunkers, similar holes should be cut of sufficient size to allow for the obstruction to the flow of the water caused by the coal. Where tunnels extend to a great height above the floor of the bunkers and are surmounted by shifting bulkheads, holes cannot be cut at a low level, and it may here be necessary to fit equalizing pipes or ducts under the tunnels. In side bunker bulkheads holes of ample size should likewise be cut and, in order to prevent the coal from running out, they should be provided with grating or wire mesh. The precautions here described serving to annul or reduce the heeling effect of longitudinal subdivision were taken by the Bureau of Construction and Repair as needed in all the troop transports under the control of the United States Navy during the war.

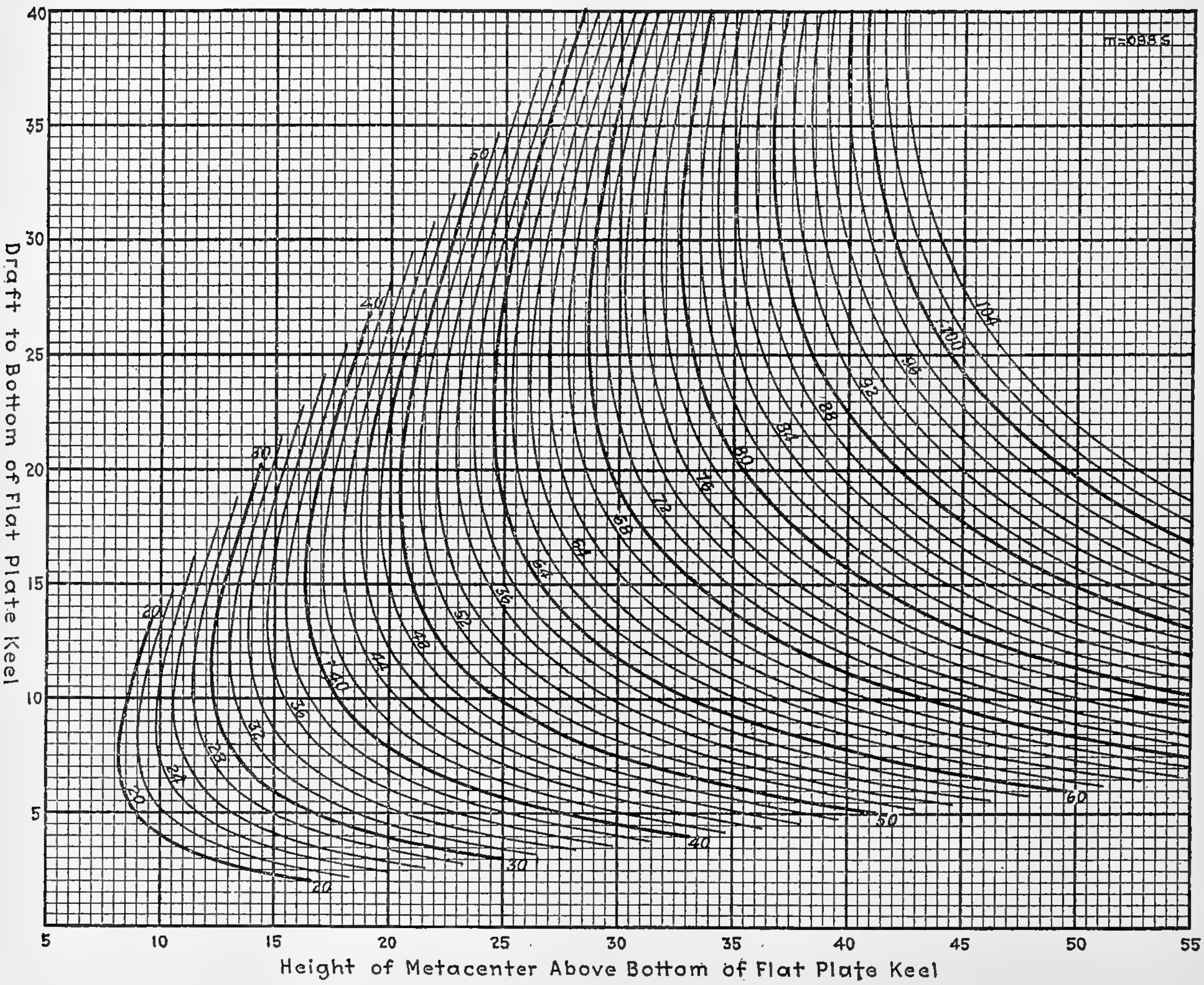
*Jahrb. Schiffb. Ges., 1916, p. 142.

DISCUSSION.

The President:—Professor Hovgaard's very admirable paper is now open for discussion, gentlemen. It is especially rich, in the opinion of the Chair, in topics for discussion.

Mr. Fred B. Webster, *Member:*—In commenting on Professor Hovgaard's excellent paper, I wish to state that the importance of this subject, as well as the method outlined, can hardly be over-emphasized. While I do not contend that other efficient methods for safeguarding the buoyancy and stability of damaged vessels cannot be devised, I believe that the subdivision of a ship into watertight compartments by transverse bulkheads is the best method so far proposed.

Metacenter Beam Chart.



For Midship Coefficient = 0.98.

Note.—For any given draught and for any desired height of metacenter above the base line, the proper beam may be selected at the intersection of the ordinate from the desired M and the abscissa from the given draught.

Undoubtedly most of you are familiar with the method of making a ship unsinkable by the use of watertight buoyancy boxes, which are inserted between the frames, under the decks between the deck beams and supplemented by tiers of boxes installed along the sides of the cargo holds—such a method, in fact, as was used on the Lucia. I understand that this method was primarily designed for use on ships where a sufficient number of transverse bulkheads did not exist, and where it would have required docking and considerable delay in fitting extra bulkheads. However, in order to make this system efficient, enough boxes would have to be installed to interfere seriously with the deadweight carrying capacity of the vessel except in those cases where the cargo consisted of coal, steel, or materials of great density.

In peace times, the subdivision of passenger vessels by Professor Hovgaard's method would appear to me to be more important than troop ships. In cargo vessels the question of sufficient length of holds for stowing piles, shapes, rails, etc., can possibly be solved, without destroying the permissible floodable length, by fitting watertight ports or doors near the tops of transverse bulkheads in line with the hatches. Such openings would permit the loading and discharging through the hatches of pieces approximately the length of the cargo hold. Even with long cargo holds it is sometimes difficult to stow material through the hatches, and I heard of one case, during the war, where some 60-foot piles that were being sent to France were sawn in two in order to get them into the ship.

Referring to ships, where such information as the design data, displacement curves, rolling or inclining experiments are not available, it is interesting to note that charts have been developed, which are to be published in the forthcoming issue of the Shipbuilding Cyclopedia, whereby the position of the transverse metacenter above base can be obtained by inspection. The charts are so plotted that the intersection of the abscissa from the desired draft with the curve of constant beam fitting the vessel will give the height of metacenter above base. The accuracy of the height of the metacenter above base can be determined by these charts to within one-tenth of a foot, providing the ships are not freaks or radically different in shape from the ordinary form.

Rear Admiral D. W. Taylor, C. C., U. S. N., *Honorary Vice-President* (Communicated) :—This paper summarizes in a general way a very large amount of work done by Professor Hovgaard when temporarily a valued member of the staff of the Bureau of Construction and Repair during the Great War. To the naval architect of the present day, given the plans of a ship, the position of her center of gravity, and plenty of time, solutions of questions of buoyancy and stability are easy. In the stress of the war, as a usual thing there were no plans, and no time. A former German vessel between 500 and 900 feet was to be fitted at a yard to carry the maximum number of troops, and must sail the moment she is so fitted, and the damages done the machinery by her former owners had been made good. Three things were done as a rule:—

First, structural additions helpful to buoyancy and stability that could be completed within the time and with the facilities available were made.

Second, holes were cut in longitudinal bulkheads to render it reasonably certain that, if a transport were sunk by a submarine, she would not first capsize.

Third, fixed ballast was added as necessary, such that, combined with the available water ballast, acceptable metacentric heights were obtainable, as explained in the paper.

It was recognized that it was not practicable, without inadmissible delay, to bring up the transports to the plane of safety against torpedo attack of a battleship, and that the main responsibility for safety of our transports rested upon our destroyers and other convoying vessels. These gave a very thorough illustration of Farragut's maxim, that the best defense is a vigorous offense, and no eastbound troop-laden transport convoyed by our Navy was touched by a torpedo. The President Lincoln, westbound, was hit by three torpedoes, rendering her sinking inevitable, but she did not capsize. Neither did the Covington, which remained afloat a night and the better part of a day after being torpedoed. The Mount Vernon was hit at about the worst possible place, at the worst possible time—so far as loss of life was concerned—just as the watches were being changed. She returned safely to port, as did the Finland.

The President:—The Chair notes the presence of a distinguished naval officer whose duty it was to fit out many of these ships during the war. Perhaps he has something to contribute to the discussion. He appears not to recognize the allusion, although I am looking straight at him.

Captain George H. Rock, C. C., U. S. N., *Member:*—I have not read Professor Hovgaard's paper. I did not get an advance copy, and I did not arrive in time this morning to hear it read—except a very small part of it at the last. I was at the Navy Yard on duty when the transports were fitted out and commissioned, and I was there when Captain Hovgaard was at the yard on several occasions making his investigations for the Bureau of Construction and Repair.

I was reminded, in listening to the latter part of his paper, of the feeling we had at the yard in getting the transports ready. It was essential, as you know, to have them sail and get the troops across, and we had to determine as to the number of troops we would send without waiting for the stability calculations. We did not have any plans of the vessels. We prepared some lines, but some of the vessels we could not dock, and therefore could not take the offsets, and were not able to draw the lines and incline them and work up their stability calculations. We had to fix on the amounts of ballast to make them safe for the live loads they were to carry, and in that work the secretary of your society, Mr. D. H. Cox, was quite prominent, and really had the greater part of the work to do.

I will just instance the way the Leviathan was handled. We settled at the yard on carrying 7,500 troops on the first trip in addition to the 500 or so officers for the army, and the 1,500 navy personnel, and requested the commanding officer to report at the end of the first trip, as to whether, in his judgment, based on her behavior at sea, the number of troops could be safely increased. The rolling tanks with which the Leviathan was fitted were not used, and we had to determine the amount of necessary fixed ballast without their use.

After the first trip, the report of the times of rolling and the behavior of the ship at sea led the Bureau of Construction and Repair to authorize an increase in the number of the troops to be carried to 10,000, and later, when we had a chance between trips, after she had made three or four round trips, we made further changes in the vessel, and the number was increased to 12,500 troops. At the same time the navy personnel was increased about 1,000 and the number of army officers was proportionately increased. On the other ships, where the numbers could not be determined in any other way, we handled the subject in the same way, and the amount of fixed ballast to be carried was settled by practical considerations and consulting with practical men around New York before the ship sailed, as we did not have time to wait for stability calculations before they had to sail with their first loads.

THE PRESIDENT:—Professor Hovgaard, will you make such comments as you desire?

PROFESSOR HOVGAARD:—I have no comments to offer.

THE PRESIDENT:—I am sure we are very much indebted to Professor Hovgaard for this concrete presentation of very valuable facts. He had an unusually fine opportunity during the war to obtain data of this character, and his paper will be very helpful to all who are interested in this subject. Its excellence is such as we always expect in papers presented by him.

One of the speakers made an allusion to the protection of the lives of passengers in peace times as perhaps being even of more importance than the protection of lives of troops in time of war. Of course a state of war means that we have to protect ourselves against conditions that fortunately do not confront us in peace times. One of the very great difficulties encountered during the war was the effect of torpedo explosion on transverse bulkheads of ships. As a matter of fact, for vessels of certain dimensions, the effective transverse subdivision which, in time of peace, would contribute greatly to the safety of passengers, would, in time of war, due to torpedo attack, have precisely the contrary effect. This resulted from the increased range of damage created by the explosive disturbance either of a mine or a torpedo. In other words, by increasing the number of transverse bulkheads in ships of certain limited dimension, you make it almost a certainty that *any* torpedo or mine striking that ship would eliminate the two-compartment safety margin by rupturing transverse bulkheads almost regardless of where it struck. That is simply a comment upon one of the incidental difficulties in war-time from which we are entirely free in times of peace.

Those who had anything to do with the deliberations of the Safety at Sea Congress, held in London in 1913 and 1914, realized very keenly the difficulty of obtaining bulkhead subdivision which will meet all the requirements of the situation both as to saving life and making an adequate provision for carrying cargo. As a matter of fact, the two subjects have to be handled more or less separately. In order to carry certain kinds of cargo, you are compelled to have a length of compartment which is not desirable, having due regard to safety under damaged conditions. The only answer to that is that you must avoid carrying any more people than you have to on cargo ships which necessitate compartments of that length. This is a slight digression from the question of buoyancy and stability, but has been brought up by comments of some of the speakers and has an indirect bearing on it.

I am sure you will permit the Chair, on behalf of the Society, to thank Commander Hovgaard for his most admirable paper.

The next paper on our program is entitled "The Application of Standardization and Graphical Methods to the Design of Cylindrical Boilers," by Mr. H. C. E. Meyer, Member.

Mr. Meyer presented the paper.

THE APPLICATION OF STANDARDIZATION AND GRAPHICAL METHODS TO THE DESIGN OF CYLINDRICAL RETURN TUBULAR BOILERS.

By Henry C. E. Meyer, Esq., Member.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

The advantages of standardization of design in quantity production are so obvious that during the recent emergency the principle of standard designs was applied with more or less success to all the many implements of war manufactured here or in the allied countries.

We have long been familiar with standard types of locomotives and automobiles, but the war was the direct cause of the development of standard ships, marine engines, liberty motors and many of the mechanical accessories of modern warfare.

In Great Britain a committee was appointed by the Institution of Naval Architects, North East Coast Institution of Engineers and Shipbuilders, Institution of Shipbuilders and Engineers in Scotland, Liverpool Engineering Society and Institute of Marine Engineers to investigate the extent to which the principle of standardization could be applied to the propelling machinery of vessels, and this committee issued a report the details of which may be found in *The Engineer* of March 21, 1919.

The most important feature of the work carried out by this committee was the development of a standard set of requirements, in so far as the structural strength of the boilers was concerned, which were recommended for adoption by the various authorities which have to do with the inspection and classification of marine boilers.

This parallels the work done in this country by the American Society of Mechanical Engineers for stationery boilers when they formulated their "Boiler Code," and is a decided step in the right direction since the existence of different rules for the same purpose evidently tends to annoyance and confusion and in many cases imposes a handicap on the builders.

Quoting from the report referred to:—"The committee thinks, however, that at present it would serve no useful end to attempt a standardization of boiler design." From this it would appear that the British committee evidently was of the opinion that the advantages to be gained, by standardizing the designs of boilers, would not justify the trouble entailed.

When it is considered that at the present time the so-called Scotch boiler still holds first place for the generation of steam on purely merchant vessels, and that in Great Britain a great number of firms have been constructing such boilers for many years, and consequently each firm has a large range of designs at its disposal, it seems but natural that considerable difficulty would be experienced in harmonizing different opinions in order to obtain uniformity of design, but in this country we have to face a somewhat different condition, and it would appear that standard designs could be readily determined upon which would both decrease the cost of production and result in more uniform performance.

While in the application of sound principles to their work the experience of many of our builders is not exceeded elsewhere, and while in the investigations concerning the propulsion of vessels possibly more valuable data have been contributed by American naval architects and engineers than by any others, the fact remains that the growth of our shipbuilding industry has been very sudden of late; and while machinery, and good machinery, has been built at points far remote from the seacoast, there are to-day many firms building engines and boilers that have not at their disposal the results from many successful installations such as are possessed by the older shipbuilders.

It would therefore appear that an effort toward the standardization of boilers would be opportune at this time, and it is the object of this paper to indicate the general direction in which this matter might be approached.

This paper is to be devoted entirely to boilers of the so-called Scotch type, which are subject to faulty design, due to the designer lacking sufficient previous data to build upon, and an effort will be made to supply data which will enable any designer to plan a boiler which will give good results.

The author has referred to the cylindrical return tubular boiler as a so-called Scotch boiler purposely, since in a publication entitled "Treatise on the Marine Boilers of the United States," by B. H. Bartol, issued in 1851, there appears on page 39 an illustration of the boilers of the S. S. Hermann, which boilers were designed by Erastus W. Smith and constructed by Mott & Ayres of New York.*

While these boilers were undoubtedly of the so-called Scotch type they were fitted with what used to be known as wet uptakes, a practice long ago abandoned when higher pressures became more common, making this type of construction unsatisfactory.

Since it appears that this boiler was originated in the United States it would only be fitting were standard designs of this type of boiler developed in this country also.

There is to-day quite a difference between boilers of the same dimensions built by different manufacturers; and, as stated before, where a new designer wishes to plan a cylindrical return-tubular boiler there is not a great deal of really definite information available to enable him to be sure of his results.

It is hoped that the data given may at least be a step in the direction of obtain-

*Due credit for the above paragraph should be given Mr. Ernest Peabody, who drew the writer's attention to the illustration referred to.

ing more uniformity of design and possibly lead to the adoption of standard designs of boilers.

A great many of the variations in design met with accomplish no useful purpose, and their elimination would prove of benefit.

It is not intended that this paper should give the impression that the author claims that the proportions outlined herein are the only ones or the best, but the data given have the merit of being based on many actual designs giving satisfactory service and do approximate the mean values of average practice.

It has been the author's experience that in many cases the specifications written for boilers were such that, had they been followed literally, the resulting boilers would have been failures, and boilers which have failed to perform satisfactorily are unfortunately not yet a thing of the past.

In order to obtain a successful boiler two factors of primary importance are more or less dependent one upon the other. These factors are heating surface and steam space. It would appear that the easiest steaming boiler is not always the one which contains the greatest amount of heating surface, and some very satisfactory boilers have contained only a comparatively small amount of heating surface for their size.

On the other hand, optimism as regards the amount of heating surface that can be crowded into a boiler is one of the primary causes resulting in designs wherein the principal characteristics are not properly balanced, and wherein conditions affecting accessibility, etc., are neglected.

We cannot sacrifice steam space indefinitely, nor can a boiler that is inaccessible have a long or satisfactory life. Steam space is undoubtedly one of the primary requisites to the proper performance of boilers, and it would appear that for ocean-going vessels an average practice is to keep the center of the upper row of tubes at one-sixth of the diameter of the boiler above the center line, whereas for coastwise or harbor vessels this distance is approximately one-fifth of the diameter, and these proportions have been rigidly adhered to in preparing the data for this paper.

Only two types of Scotch boilers have been considered, *i. e.*, those with separate combustion chambers and those with a common combustion chamber, and while there are other variations they are not by any means common.

The separate combustion chamber type is more particularly suitable for ocean-going service, whereas the common combustion chamber type is used principally for coastwise and tugboat service. When forced draft is used the separate combustion chamber becomes almost essential.

Perhaps one of the best reasons for adhering to the separate combustion chamber type of boiler for ocean-going vessels lies in the fact that the interior of this type of boiler is so readily accessible for cleaning purposes, and there can be no doubt that a boiler which is examined and thoroughly cleaned at regular intervals is going to give the best service and have the longest life.

It is not the intention to discuss in this paper the methods for arriving at the heating and grate surface required to develop a given power, but rather to determine upon the dimensions of a boiler to give a predetermined heating surface. While the determination of the heating and grate surface required ought to be based upon actual steam consumption it is to be regretted that there exists very little published data, determined by actual trial, as to the real consumption of steam by different types of marine propelling machinery, or as to the actual evaporation of water by boilers of the type under consideration.

On the other hand, there are so many successful vessels in operation to-day that the determination of the heating and grate surface is fairly simple, even though methods are used which are perhaps not quite as scientific as they might be; and while the possession of more accurate data might enable the designer to more closely approximate his requirements with the actual performance of the boilers, in merchant vessels reasonable excess of boiler power can never be considered a fault.

The life of a boiler constantly forced to the limit is bound to be shorter than that of a boiler which readily keeps the machinery supplied with steam at a constant pressure, and it would also appear that, where fires are forced, the losses due to imperfect combustion would be greater.

Another very probable cause for the failure of boilers to steam properly lies in the fact that if the designer is limited to diameter of boiler he is apt to overestimate the amount of heating surface that can be placed in a boiler of a given size; and while there is no doubt that the heating surface which is contained in most boilers could be considerably increased by reducing the available steam space or by crowding the tubes, it is very much to be doubted whether the boilers would be improved thereby.

The writer has come across cases where the heating surface had been increased by keeping the tubes very close to the furnaces, and while the calculated heating surface was thereby increased it is quite possible that the elimination of all the tubes which were in close proximity to the furnaces would not have had any appreciable effect on the steaming qualities of the boiler.

One of the causes of the great differences existing in designs lies in the fact that prior to the war, when the individual shipyards of this country were engaged on many different types of vessels at the same time, ranging from ferry-boats to big passenger liners, the conditions were not favorable to the development of standard designs.

A design for a 10-foot-diameter boiler might be followed by another for a 15-foot boiler, and this again by a design for a 13-foot boiler.

Each case being developed upon its own merits, it was to be expected that regular progression of the various proportions of different boilers would be given but scant consideration, with the result that in some cases boilers of varying diameters have had the same heating surface and diameter of furnace.

With this condition in mind the writer at one time made a series of designs progressing at the rate of 3 inches in diameter of boiler, which designs proved that a regular progression of dimensions could be maintained and that the resultant designs would have well-balanced proportions.

The designs referred to were limited to boilers with separate combustion chambers for forced draft, and boilers with common combustion chambers for natural draft, all with three or four furnaces, but did not cover two-furnace boilers.

In the accompanying diagrams, boilers with separate combustion chambers for natural draft have also been included, and the diagrams for boilers with common combustion chambers have been extended to include two-furnace boilers.

While some two-furnace boilers are fitted with separate combustion chambers, these boilers generally are small, as in the case of small vessels for coastwise or harbor service or of donkey boilers, and it would seem that a common combustion chamber type would be equally acceptable in such cases and would be considerably cheaper to construct.

FURNACES.

The diameter of furnace that can be used conveniently in a boiler of a given size is a question that may be open to considerable discussion. It is, however, one of the first and most important features of design, since a furnace too large will result in either unduly crowding the tubes and reducing the steam space available or having an insufficient amount of heating surface and area through tubes, whereas a furnace too small in diameter will result in poor combustion and insufficient grate surface.

In determining upon a standard design, therefore, the diameter of the furnace should be a prime factor, and Plate 57 gives furnace diameters which not only are in accord with average practice but which increase systematically in proportion to the diameter of the boiler.

Attention is again drawn to the fact that there seems to be considerable difference of opinion on this subject, some builders using smaller, others larger furnaces, but the dimensions given will ensure a well-balanced design when all the other features have been taken into account, such as the ratios between heating and grate surface and between area through tubes and grate surface.

It should be remembered that in Plate 57 the diameters of furnaces given are the extreme outside diameters and not the effective diameters.

TUBES.

While there are many different sizes of tubes used the most common practice seems to be to use 3-inch O. D. tubes for natural and 2½-inch O. D. tubes for forced draft.

It is evident that by the use of smaller tubes the heating surface in a boiler can be increased, but there seems to be no really sound reason why in a series of standard designs of boilers the above diameters could not be adhered to.

Plate 58 gives the number of tubes for each size of boiler.

At this point it is well to note that the number of tubes which can be placed in boilers of a standard type and with standard characteristics will not give quite as regular a curve as plotted on the above diagrams.

The spots indicated on the diagrams give the actual number in the case of some standard designs actually worked out, and it will be noted that at various points a step occurs in the curve. This step is due to the fact that when the point is reached where, due to the increase in size of the boiler, it becomes possible to add another horizontal row of tubes, the number of tubes is increased somewhat rapidly.

As regards the length of the tubes, this is of course dependent upon the length of the boiler, depth of combustion chamber, and depth of water space back of the combustion chamber.

It would appear that by definitely settling upon certain desired characteristics of boilers, *i. e.*, the ratio of heating to grate surface and the ratio of area through tubes to grate surface, the lengths of a complete series of boilers could be kept down to very few different dimensions, which would result in the possibility of keeping the tubes of certain standard lengths, possibly only three or four lengths being required, which would enable both the manufacturers and boiler-makers to keep stocks of tubes for ready use.

It would also have the further advantage that the lengths of furnace could be kept standard, and there is no reason why these should not be made to standard dimensions throughout, which again would make possible the standardization of furnace fittings.

Anyone familiar with repair-shop practice knows that unless each vessel is provided with patterns for the side bars for each furnace, whenever it becomes necessary to renew same, new patterns have to be made and fitted to each furnace, which in itself constitutes a serious economic waste.

Lastly, the boilermaker would be benefited by the fact that all longitudinal stays could be made of standard dimensions.

COMBUSTION CHAMBERS.

The depth of combustion chambers used in connection with these diagrams has been plotted on Plate 57. The proportions given are quite sufficient for coal burning and generally should be sufficient when burning oil, although some designers prefer to use a greater depth in the latter case.

It should be kept in mind that the dimensions given are inside dimensions, and the actual depth used should be stated in even inches, which will somewhat simplify the calculations for allowable pressures on plates, stays and girders, etc.

HEATING AND GRATE SURFACE.

Based upon the dimensions for furnaces and combustion chambers given by Plate 57 and upon the number of tubes given by Plate 58, the diagrams for heating surface, grate surface and area through tubes have been constructed.

By reference to Plates 59, 60 and 61 it is possible to quickly estimate how much heating surface may be expected from a boiler of certain dimensions or, knowing the heating surface required, what size boiler is needed. In this connection the writer would again point to the comparative scarcity of published data on the performance and efficiency of Scotch boilers.

The different authorities on the subject give figures for heating surface required that vary widely, and there is no doubt that the tendency to keep the boilers too small is somewhat of a temptation when costs have to be kept down.

After all, the most reliable data for designing boilers are not the result of builders' trials when all conditions are favorable, but rather the average result of actual long voyages of vessels which are no longer new but which have been in operation some time.

There is no doubt but that it might mean some expense to the shipowner to obtain these results, as in many vessels the apparatus for indicating the engines is taken down after the builders' trials and never used again. On the other hand, a careful investigation of the performance of a vessel not giving satisfactory service would probably result in the adverse conditions being eliminated, with consequent advantage to the owners.

As a guide to what can be accomplished the writer offers the following data taken from actual performance during long voyages of a tramp steamer:—

Boilers.

No.	Two
Type	S. E. cyl. ret. tubular
Draft	Howdens
Heating surface total,	4,464 square feet
Grate surface total	104 square feet
$\frac{\text{H. S.}}{\text{G. S.}}$	44 square feet

Auxiliaries.

Air pump	Lever driven
Circulating pump	Lever driven
Feed pump	Lever driven
Bilge pumps	Lever driven
Blower engine,	Independent reciprocating
Electric generator,	Independent reciprocating
Steering engine	Independent

Engines.

Cylinders	24-60-67
Stroke	45
Average revolutions	65
Pressure at throttle	195
I. H. P.	1,600

Consumption of Fuel.

Type of coal	Welsh
Coal in 24 hours	24 tons
I. H. P. per sq. ft. grate surface	15.4
Coal per I. H. P.	1.4 lbs.
Air pressure at blower	1⅜-inch

The above figures are given as the writer is thoroughly familiar with the particular case and knows that this vessel gave very satisfactory results on voyages of from eighteen to twenty-one days' duration from port to port.

In addition to the diagrams giving the characteristics of designs of a series of boilers, the writer has added diagrams for obtaining the weight of material entering into construction of a boiler and also the weight of water.

The weights of materials are gross weights and do not include fittings.

In examining the diagrams it is of course impossible for anyone not familiar with the designs whereon same are based to decide at a glance whether these diagrams will give proportions that are reasonable. For this purpose the writer has prepared Plates 65, 66 and 67, which boilers were designed from data taken direct from the diagrams given. These designs do not present any unusual features, and in preparing them no difficulty was experienced in obtaining the proportions which the charts predicted.

The arrangement of tubes was kept as simple as possible in order to avoid unnecessary complication of the smoke boxes since the few additional tubes that might have been added would only be of doubtful value anyway.

The diagrams may be used very readily for making a table of standard boilers, and it is to be hoped that the data as given may prove of some benefit to the designer who has not at his disposal a number of designs which have proven to be successful.

For the designer who has already a great number of designs available it may follow that the methods outlined in this paper may be of benefit in coordinating his various dimensions of boilers.

The following example of the use of the diagrams may possibly demonstrate their value somewhat more clearly:—

Let us suppose that it is desired to design a boiler having the following characteristics:—

Type.	*Separate combustion chamber.*
Heating surface	2,200 square feet.
Grate surface $= \frac{\text{H. S.}}{34}$	66.3 square feet.
Area through tubes not less than...........	1,800 square inches.
Grates not longer than....................	6′ 3″
Draft	Natural.

Since the length of grate bar is fixed, the grate area will be the determining feature as to the diameter of the boiler.

By reference to Plate 60 it will be seen that bars 6 feet 3 inches long will give a grate surface of 67.5 square feet in the case of a boiler 15 feet diameter, *i. e.*, the diameter of furnaces appropriate to this diameter of boiler is such as to give this amount of grate surface.

The same diagram also shows that if this boiler is made 11 feet long we will have 2,220 square feet of heating surface and an area through the tubes of 1,890 square inches.

The diameter of the furnace can next be obtained from Plate 57 and is found to be 3 feet 11 inches outside, while the depth of combustion chamber would be 33 inches.

Plate 58 gives the number of tubes as 320.

Allowing a water space of 9 inches at the back of the combustion chamber and 3 inches for the total thickness of heads, and combustion chamber back and tube sheets, the length of the tubes between sheets will be 7 feet 3 inches.

The weight of water obtained from Plate 64 will be 20.6 long tons.

The above results may now be compared with the results actually obtained in the design illustrated by Plate 67, which results were calculated.

	Estimated.	*Actual.*
Diameter	15′ 0″	15′ 0″
Length	11′ 0″	11′ 0″
Diameter of furnaces	3′ 11″	3′ 11″
Number of tubes	320	327
Eff. length of tubes	7′ 3″	7′ 3″
Heating surface	2,220	2,288
Grate surface	67.5	67.5
Area through tubes	1,890	1,962
Depth of combustion chamber	2′ 9″	2′ 9″
Weight of boiler	102.500 lbs.	
Weight of water	23.2 tons	

It will be seen that the design in Plate 67 closely approximates the desired characteristics and that specifications can be quickly prepared by this method.

DISCUSSION.

The President:—Mr. Meyer's paper, "The Application of Standardization and Graphical Methods to the Design of Cylindrical Boilers," so ably presented by the author, is now before you. Is there any discussion on the paper?

Mr. E. A. Stevens, Jr., *Member:*—Mr. Meyer should be congratulated on devoting the time and attention to turning out a series of charts such as he shows in this paper. More than once the boilers of a ship have been ruined by trying to crowd in too much heating surface, due to lack of proper information as to the size required, to obtain the required heating surface. Many boilers would be in much better condition today if they had had less heating surface.

There is one question I would like to ask Mr. Meyer which I do not see stated in his paper. What is the pitch of the tubes, and if the pitch is determined according to the diameter or by the distance in between the tubes? This is important not only for proper circulation, but for cleaning as well. The curves showing the weight of water and the weight of boilers would also be useful in getting out preliminary calculations for a vessel.

Mr. Meyer:—In regard to Mr. Stevens' point, as to the pitch of the tubes, I am glad he mentioned that. All the diagrams were based on a standard distance between tubes, that is to say, vertically, the distance from the outside of one tube to another is 1 inch, and 1¼ inches horizontally. That point has been brought to my attention on several occasions.

In an endeavor to crowd in a large amount of heating surface, the designers have often gone to work and spaced the tubes closely together. In some cases I have seen boilers where the distance between the tubes was only 13/16 inch. I remember one case where some people who were designing a boiler, stationary engineers, not accustomed to marine practice, had as low as ½ inch. The best practice is to keep the distance between the tubes 1 inch vertically and 1¼ inches horizontally, both for the purpose of cleaning the tubes and getting proper circulation.

As regards the weights of boilers, the diagram of the weights is based on something like 300 calculations of individual boilers, and I am safe in saying these are correct within 1 or 2 per cent, unless there are exceptional features in the design of the boiler.

Commander Stevenson Taylor, *Past President*:—I have not very much to say on this paper, except to express my appreciation of the work which has been done towards standardization, and that brings to my mind the different points of view as to the necessity of standardization. It is but a very few years ago that one of the most distinguished members of our Society, in speaking on the subject of making standardized ships, said: "You might just as well have wives standard for all men as to have ships standard for owners." Of course the developments of the last two years have perhaps changed his mind, though I think not. I have not discussed the question with him recently.

Mr. Stevens has brought to mind an important question as to the design—that is, the space between the tubes—and I notice that the plans on Plates 65 and 66 show no occasional extra water-spacing between the tubes such as it has been my practice for years to make. It seems to me that the design on Plate 67, in that respect, would be a better design because such extra water-spaces furnish a greater opportunity for circulation of water.

I would like to ask Mr. Meyer to express his opinion on that subject in his concluding remarks. Perhaps he has had experience that warrants putting so many tubes together, without extra spacing, as he shows on Plates 65 and 66.

May I express my appreciation, sir, of the care with which Mr. Meyer has prepared this paper, which will be very useful for quickly determining at least the main points of boilers in designing them for any required purpose?

The President:—Does Mr. Meyer desire to present any further remarks?

Mr. Meyer:—The point that Mr. Taylor has just brought up regarding single-combustion chamber boilers, *i. e.*, the spacing of the tubes above the furnace, is interesting, and I am aware that many of the designers do space the tubes above the center of the furnace in single-combustion chamber boilers, instead of 3½ to 4¼, as much as from 7 to 9 inches apart, and some people do consider this good practice. I myself think there is a good deal to it, because naturally the crown of the furnace is the most efficient heating surface, and you will be more likely to obtain definite circulation by providing space for the water to go through. However, in the designs with which I have been familiar we did not do that. I have in mind a certain boiler, with a single-combustion chamber, which is 16 feet in diameter, with three furnaces, and it has a tremendous number of 2¾ tubes and does not have any of these large spaces which have been referred to. However, these diagrams I do not think can be affected very much by the adoption of this practice, provided you do not make the space too great, because by spreading these tubes slightly above the center of the furnace, it would merely mean the elimination of three or four tubes on the circumference of the boiler, which could readily be taken care of in estimating your heating surface from the diagrams. Personally, I think it is a good thing to do.

Mr. Edward C. Gillette, *Member* (Communicated) :—The subject has been ably handled by Mr. Meyer and covers a number of points which have not heretofore appeared clearly in print, and which are of invaluable assistance to the designer in cutting corners and checking his work.

I have no comments to make except that, as the shipping world is now resorting to oil as fuel, it would appear that additions to your present data covering oil-burning boilers would be very interesting and advantageous. In this connection it appears, from my experience, that the question of added heating surface in furnaces with the omission of grate bars, etc., the use of separate *vs.* one combustion chamber common to all furnaces; size of tubes or the use of retarders in boilers converted from coal to oil fuel, and the size of stacks and heights of same, etc., should be given some consideration.

With reference to the use of one combustion chamber common to all furnaces *vs.* separate chambers, there are advantages and disadvantages in both, but it is thought that a boiler fitted for burning oil, when under "banked fires," *i. e.,* one burner in operation, the one common chamber is preferable in that the heat is more evenly distributed, tending to equalize in the chamber and all furnaces, thus eliminating expansion and contraction strains, which would be more pronounced with separate chambers.

The President:—Commander Stevenson Taylor expressed the opinion of the Chair with reference to Mr. Meyer's paper. It is a record of interesting and valuable data, and I am sure you will permit the Chair to extend the Society's thanks to Mr. Meyer for his painstaking care in the preparation of this paper and its presentation.

The next paper is entitled "New Developments in High Vacuum Apparatus," by Mr. G. L. Kothny, Member.

Mr. Kothny presented the paper.

NEW DEVELOPMENTS IN HIGH VACUUM APPARATUS.

By G. L. Kothny, Esq., Member.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

The adoption of the steam turbine for marine propulsion has brought about many changes in marine engineering, which have become necessary in order to fully realize the many economical advantages obtainable. These changes affected particularly the vacuum producing apparatus. Steam turbines require the highest possible vacuum for highest efficiency and economy. The importance of this has been realized, and considerable research work has been carried out and many developments have been made in relation to the condensation of steam and the extraction of air.

The principles of surface condenser design have been dealt with in several papers to various societies during the last few years, many of which have shown a thorough knowledge of the physical laws and practical considerations governing the subject. However, very little has been said in these papers about the design and development of the air pump, which is the most important part of a vacuum-producing apparatus.

It is intended in the following to record some developments which have been made during the last few years in the extraction of air from marine condensers.

The recognized limits of vacuum for well-designed marine turbine surface condensers based upon the inlet temperature of the circulating water are as follows:—

1½ inch Hg. absolute with sea water at 60° F.
2-inch Hg. absolute with sea water at 70° F.
2.4-inch Hg. absolute with sea water at 80° F.
2.75-inch Hg. absolute with sea water at 85° F.

When estimating the correct size of air pump to use, the chief factor to bear in mind is the normal air leakage which the pump must handle. No definite rule can be laid down for the amount of air leakage, as it depends on the size and character of the installation. In small installations this is relatively higher than in large ones. Also in condenser installations where a large number of auxiliaries, particularly those located on deck, are exhausted into the main condenser, the air leakage is greater than in a normal installation, and a larger air pump will be required.

Extensive investigations have shown that the normal air leakage for marine surface condenser installations can be limited to the amount shown in Plate 68, provided that any auxiliaries exhausting into the main condenser do so at a pressure slightly above atmosphere. The curve illustrated in Plate 68 is based on the total amount of steam which the condenser receives. It also includes the air contained in the exhaust steam.

The air received from the condenser is always fully saturated with vapor, the amount of which depends upon the temperature and absolute pressure. The relations between water vapor and air at different absolute pressures and temperatures based on Dalton's Law are illustrated in Plate 69.

In determining the air-handling capacity of the pump, the amount of vapor should be deducted from these curves and added to the normal air leakage given in Plate 68.

In looking over Plate 69 it will be seen that at a constant absolute pressure the amount of vapor decreases with the drop in temperature. It is therefore desirable to remove the mixture at the lowest possible temperature in order to decrease the size of air pump. This fact has been recognized and has led to the adoption of the dry system in which the air and the condensate are removed separately.

The dry system permits the removal of the air and vapors at the lowest possible temperature by a dry air pump; the condensate is taken away at a higher temperature by either a turbine-driven or reciprocating condensate pump. Since the adoption of the dry system, many developments have been made in dry air pumps, the most notable being the steam air ejector, which, due to its many advantages, has been extensively used during the last three years.

The use of steam jets for the removal of air has been known for over fifty years. As early as 1868 a patent was granted for a steam-jet air pump. But this type of pump has never been developed to produce high vacua commercially until recently. This was due to the lack of the correct knowledge of the properties of steam and the want of interest in high vacuum. During the years 1900 to 1910 the properties of steam have been more accurately determined by several prominent scientists who made it possible to carry out experiments for the improvement of steam air ejectors. At the same time the demand for high vacuum became more pronounced, due to the adoption of the steam turbine.

The steam air ejector is a compressor in which the air to be compressed is entrained by one or more steam jets, which move through the entrainment space at a very high velocity. The entrainment is made by friction. The kinetic energy, originating from the steam jets, and contained in the mixture of air and steam after entrainment, is transformed into pressure in a channel called the diffuser.

Referring to Fig. 3, Plate 70, air enters through opening *A* into the entrainment chamber *E;* live steam expands through one or more nozzles whereby its static energy, due to its pressure, is transformed into kinetic energy. This transformation causes the steam to leave the nozzle with a velocity ranging from 3,000 to 4,000 feet per second. The steam jets *J,* while passing through the entrainment chamber *E,* entrain the air and give up part of their kinetic energy to the same. The outlets of the nozzle or nozzles are arranged opposite the opening of the diffuser, and the mixture of steam and air enters the diffuser at a somewhat lower velocity than that of the steam jets leaving the nozzles. The mixture is gradually brought practically to a rest in the diffuser D, thus increasing its static pressure to or slightly above atmosphere.

The efficiency of a steam air ejector depends upon the magnitudes of three losses:—

(a) The friction losses in the expansion nozzle.
(b) The impact losses during entrainment.
(c) The losses in the diffuser.

Due to the development of steam turbines the nozzle design has approached nearly the highest degree of efficiency possible, and the friction losses in a well-designed steam nozzle are seldom more than 10 per cent (see H. Frederic and Kemble, Transactions of the American Society of Mechanical Engineers, 1909). These losses depend mostly on the length of the nozzle and the condition of the inner surface.

The loss due to impact is influenced by the relation that the momentum of the steam, plus the momentum of the air before the impact, must equal the momentum of the co-mingled steam and air after impact.

If we designate the amount of live steam to be G_1 pounds, the amount of air entrained G_2 pounds, the velocity of the live steam at the outlet of the nozzles V_1, the velocity of the air V_2, and the velocity of the mixture V_3, we can write the following equation:—

$$G_1 \times V_1^2 + G_2 \times V_2^2 = V_3^2 \times (G_1 + G_2).$$

The impact loss can be minimized by letting the air acquire a higher velocity before combining with the steam jets. This, however, would make it necessary to produce a smaller absolute pressure in the entrainment space and to provide an annular air nozzle ahead of the same. As a result the pressure rise in the diffuser would have to be greater than otherwise, and consequently the loss in the diffuser will be increased. Careful experiments and calculations have shown a certain range of conditions within which the sum of all the losses is a minimum. The under-expansion and arrangement of an air nozzle prior to the entrainment has been found to be of advantage only for very small compression ratios, which, however, are not used in steam air ejectors producing high vacuum.

The losses in the diffuser are considerably larger than those in the nozzle. They range from 25 to 50 per cent and are influenced by the length and the shape of the diffuser, as well as by the conditions of the inner surface.

The ratio of compression is the ratio between the absolute pressure of the air at the air intake and that at the air discharge.

Since turbine condensers require vacua of 2-inch Hg. absolute or less, the ratio of compression has to be 1 to 15 or more. While it is possible to obtain such a high compression ratio in a single stage, carefully made experiments have shown the inadvisability of using such a high compression ratio single-stage ejector for practical purposes. The following conclusion will illustrate why:—

Assumed that a single-stage air ejector is designed to produce a vacuum of 2-inch Hg. absolute. When starting this ejector, the absolute pressure at the air intake is the same as that at the air discharge, namely, atmospheric pressure. The steam will leave the nozzle with considerably less velocity than that which it would obtain under normal operation. The efficiency of the nozzle will be decreased, and the entrainment surface (outer surface of the steam jet) will be decreased. The air to be entrained is in a less rarefied state, hence the loss due to the impact during entrainment is also greater. Consequently the velocity of the mixture is smaller and its volume is larger. As this mixture has to pass through the throat of the diffuser, the sectional area of the diffuser would have to be enlarged for starting conditions. If this is not done, starting will become impossible. Assumed that the steam jet after leaving the nozzle does not touch the diffuser walls, but leaves an annular opening or channel between the outer surface of the steam jet and the inner walls of the diffuser through which the air is entrained (as shown in Fig. 4, Plate 70), a vacuum will be produced which will gradually increase with the decrease of the losses. The velocity of the steam will increase with the vacuum. This in turn will retard the enlargement of the diameter of the steam jet. The increasing difference of pressure in the diffuser will recede farther towards the diffuser throat. And if the designed vacuum is obtained before the steam jet touches the diffuser wall, the ejector will start spontaneously.

If, however, the jet touches the diffuser walls before the designed vacuum is obtained, the entrained air cannot pass. It will re-circulate, as illustrated by the arrows in Fig. 5, Plate 70, and the ejector will never start. This will happen if a single-stage ejector is designed for 2-inch Hg., because its diffuser throat area will be entirely too small for starting conditions.

If increased in sectional area to overcome those, the section will be too large to be filled out with the co-mingled steam and air at 2-inch absolute, and atmospheric air will rush back into the entrainment space and the desired vacuum will not be obtained.

To overcome this difficulty such a single-stage high-compression ratio ejector would have to be provided with either a flexible diffuser or special arrangements for starting.

In all ejector designs, whether for low or high compression ratio, the above facts have to be given careful consideration. The diffuser throat is generally enlarged over the required size for design conditions. This enlargement represents a compromise between diffuser losses and the self-starting capacity. The latter is more important than high efficiency. In practical installations vacuum conditions vary, and lack of this self-starting characteristic will cause the ejector to stop working, which may have serious consequences. Experiments have shown that the maximum ratio of compression for a single-stage ejector working in connection with condensers should not exceed 1 to 7. In other words, when it is desired to have more than 25½-inch vacuum with 30-inch barometer, it is advisable to arrange two ejectors working in series and divide the ratio of compression. All high-vacuum ejectors used with marine-turbine condensers consist of two stages—the first stage, which is connected to the air suction of the condenser and which exhausts into the second stage at a pressure varying from 4 to 10-inch Hg. absolute; the second stage, which compresses the mixture of steam, air and vapor received from the first stage to a pressure slightly above atmosphere. The steam used in both stages, while passing through the ejector, gives up its energy but does not lose its heat, with the exception of some small losses due to radiation. The exhaust from the ejector is discharged into the feed tank. The heat in the steam is by this means transmitted directly to the boiler feed with practically no loss.

The characteristic of the performance of the steam-air ejector is illustrated in Plate 71. Curve A shows the air-handling capacities of a two-stage ejector designed for 2 inches absolute at different absolute pressure, when exhausting dry air and when being supplied with a constant quantity of dry saturated steam of constant pressure, and when exhausting against a constant back pressure of ½-pound gauge. It will be noted that the air ejector is capable of producing an absolute pressure of 0.25-inch Hg. at dead end (with the air suction blanked off). The capacity increases steadily with the increase of absolute pressure.

The actual dry-air handling capacity of the second stage of the same ejector working under similar conditions is illustrated in curve B. The steam consumption of the second stage is, of course, less than the total steam consumption of the two stages.

Curve C, shown in dot-and-dash lines, represents the air-handling capacity of the second stage based on the total steam consumption of the two-stage ejector.

Comparing these curves it will be noted that at 5-inch Hg. absolute the second stage alone handles as much air as both stages together, and that from this point the capacity of the second stage increases over that shown in curve A. This is due to the fact that, when both stages are working, the second stage also compresses the steam discharged from the first stage. One is accordingly led to the conclusion that it is advisable to shut off the steam supply to the first stage at about 5-inch Hg. absolute to save the live steam used in the same. Curve C also indicates that at about 4.5-inch Hg. absolute or more a single-stage air ejector should be used, provided it can be designed self-starting.

In connection with Plate 71, it must not be assumed that the curves indicated are in every sense absolute. They depend entirely upon the ratio between the compression ratios of the first stage and those of the second stage. The curves shown are those of an ejector which has been extensively used in marine installations.

The main factors (excluding those referring to the design) controlling the performance of a steam-air ejector are:—

(*a*) The quality of the live steam supplied.
(*b*) The pressure of the live steam supplied.
(*c*) The back pressure at the discharge.
(*d*) The condition of air handled.

The live steam supplied to the steam-air ejector should be dry saturated or slightly superheated. If less than 98 per cent saturated, the losses will rapidly increase and the stability of operation will suffer. The presence of water in the steam can easily be detected by a whistling noise and a sudden drop of pressure of the steam at the pressure gauge arranged ahead of the inlet to the nozzle. If the possibility of obtaining wet steam exists, a steam separator should be provided.

High superheated steam, while good for steam turbines, does not offer any advantages for the steam-air ejector. Its compression in the diffuser becomes difficult, and this counterbalances the benefit obtained from its use in the nozzles.

The necessary minimum pressure of the live steam at the inlet to the nozzles is a function of the compression ratio the ejector has to produce. The kinetic energy of the steam, after expansion through the nozzle, has to be large enough to compress the mixture of steam and air. The minimum steam pressure employed varies from 100 to 110 pounds gauge. The influence of the variation in steam pressure on a two-stage air ejector designed for 110 pounds gauge removing a constant quantity of air against a constant back pressure is shown in Plate 72. It will be noted that even low steam pressures will produce some results.

The effect of the back pressure upon the performance is illustrated in Plate 73, several curves for different live steam pressures being shown. All are based upon a constant air-handling capacity. These curves also evidence the possibility of overcoming a higher back pressure by an increase in the live steam pressure.

The effect of the conditions of the air entrained is based upon the following considerations. Air saturated with water vapors weighs considerably more than dry air. Since we can assume that the ejector will handle the same total weight whether the air is dry or saturated, the application of the Dalton Law (see Plate 69) will enable the determination of the respective quantities. Referring to curve A in Plate 71, at 2-inch Hg. absolute, 77 pounds of dry air were handled when drawn directly from the atmosphere. Assume now that fully saturated air is to be handled at the same absolute pressure and that this air has a temperature of 80° F. The curves of Plate 69 indicate the presence of 0.67 pound of water vapor to each pound of dry air in 1.66 pounds mixture. Hence the ejector would theoretically handle $\frac{77}{1.67} = 46$ pounds of dry air. Careful experiments have shown the amounts actually handled to be less than the calculated theoretical values. This is partly due to small globules of water suspended in the saturated air being carried into the ejector. An air ejector handling a mixture containing a considerable amount of vapor also has greater impact losses during entrainment and greater losses in the diffuser. The weight of mixture handled by the ejector becomes less as the proportion of vapor to dry air increases.

Since the air withdrawn from a condenser is always saturated with water vapor, the above-mentioned condition should receive careful consideration when selecting the proper size of an air ejector for a condenser installation.

Having thus briefly indicated the principles of operation, as well as the most important factors in steam air ejector design, several types of ejectors will be described which have been in successful operation in marine condenser installations over two years and thereby have proven their fitness for marine service. Only two types in a variety of sizes have so far been used in a large fleet of merchant marine and navy vessels.

These are known as the "Radojet" and the Le Blanc air ejector. Both are of the two-stage type, the stages being connected in series. A cross-section through the Radojet is shown in Plate 74. Plate 75 shows a photograph of half a Radojet cut through its vertical center line.

Live steam is delivered from a source not shown through opening *L* through strainer screen 1, pipe 2, auxiliary steam valve 3, strainer screen 4, expansion nozzles 5, across suction chamber 6, of the first stage ejector, which is in communication with the condenser through the suction opening *S*.

The steam expands in the nozzles, leaving the same with a very high velocity and, while passing across suction chamber 6, entrains the air and vapors to be compressed.

The mixture passes into the diffuser 7, from where it is discharged at higher absolute pressure than that of the air entering at *S* into a double passage 8 communicating with the suction chambers 9 of the second stage. These two suction chambers 9 are annular, giving the co-mingled fluid a large entrainment surface.

Steam is simultaneously delivered through the strainer screen 1 into passage 10, which communicates with the annular expansion nozzle formed between nozzle 11 and nozzle point 12. Nozzle point 12 may be adjusted toward or away from disc 11 by the adjusting screw 13 which forms part of it, to vary the cross-section of the nozzle passage, thereby changing the expansion ratio of the steam.

The steam delivered by the annular nozzle 11 expands between the same and nozzle point 12, leaving it as a jet of high velocity in the form of an annular sheet. In passing across the section chambers 9, it entrains the co-mingled air and steam received from the first stage and carries the mixture into the annular diffuser 14, thereby compressing it to slightly above atmospheric pressure and discharging it into casing 15 which has the discharge opening *D*.

The steam nozzles of the first stage are bronze, that of the second stage is of monel metal, and the nozzle point of special steel. The diffusers are bronze. In the smaller sizes, the diffusers form part of the casing, while in the larger sizes the diffusers are secured to a cast-iron casing by bolts, forming a metal ground joint with the casing.

The strainers ahead of the nozzles protect them from becoming clogged by foreign substance. They are easily removed and are made of perforated monel metal or monel metal wire gauze. The openings in the strainers are smaller than the throat of the nozzles.

The particular features of this type of air ejector are:—

1. The use of a radial jet (from which the Radojet has derived its name), which has a greater penetrating force and larger entrainment surface than that of a cylindrical jet of equal length. Consequently better efficiencies are obtained.

2. The possibility of adjusting the expansion ratio of the live steam from the outside without changing the steam consumption. Referring to Plate 74, it will be seen that the throat is located in nozzle 11 and that any movement of the disc to or fro will not change the cross-sectional area of the throat and therefore not affect the steam consumption.

A cross-section through the Le Blanc air ejector is shown in Plate 76. It has one DeLaval nozzle in the first stage, and multiple DeLaval nozzles are arranged close to the inside of the diffuser walls in the second stage. It is claimed that the losses due to the impact in the second stage are reduced since the mixture discharged from the first stage enters the entrainment chamber of the second stage with greater velocity and in a direction nearly parallel to the steam jets of the second stage. The throat of the diffuser in the second stage also has an opening 2 connected with the atmosphere. The object of this opening is to permit the air thus drawn in to form a flexible diffuser wall. At starting and at low vacua the pressure at the diffuser throat will be atmospheric. With the increase of vacuum this pressure will be lowered and the inrushing air will partly fill up the diffuser throat, thereby decreasing the effective area. While this arrangement conforms with the principle of self-starting as outlined above, it adds to the work of the second stage, as the air thus admitted has to be compressed and ejected.

There are also a number of other steam-air ejectors on the market, none of which to the writer's knowledge have been in operation for marine service. All of these have an inter-condenser between the first and second stages. The object of this inter-condenser is to reduce the work of the second stage by condensing the steam discharged from the first stage, thereby decreasing the total steam consumption 30 to 40 per cent. While this scheme may look very attractive at first, a closer investigation will show that there is no advantage in using an inter-condenser between stages in marine installations.

On board a ship it is necessary to install a surface inter-condenser. The heat contained in the steam of the first stage is entirely lost, as it is absorbed by the circulating water. Provision has to be made for removing the condensate from the inter-condenser. Circulating water has to be supplied to the inter-condenser, which means additional work. The weight of the inter-condenser air ejector with its necessary piping is about eight times that of the straight two-stage ejector. The space required is five times as large and the arrangement of the piping is complicated. The increased weight and the larger space required mean a slight increase in propelling power. Space and weight also make it difficult to provide two or three steam-air ejectors with inter-condensers working in parallel. Such an arrangement can easily be made with a steam-air ejector without an inter-condenser and has the advantage of increasing the flexibility and economy during light loads.

The gain in steam consumption by using the inter-condenser is more than counterbalanced by all the disadvantages mentioned. The arrangement with an inter-condenser also lacks the simplicity which a two-stage air ejector installation without inter-condenser possesses.

According to the opinion of numerous marine operating engineers, simplicity is of greater practical value than the saving of a few pounds of live steam.

Until the appearance on the market of the steam-air ejector the twin-beam air pump quite monopolized the field. Therefore it will be interesting to compare the two types.

Considering first the comparative steam consumptions based on equal air and condensate handling capacity it will be found that the steam consumption of the air ejector, plus that of a turbine-driven condensate pump, is lower than that of the twin-beam air pump, even if the latter is arranged with a dry and a wet cylinder.

The weight and space requirements are also decisively in favor of the air ejector outfit. Plate 77 illustrates a set of either type of equal air and condensate handling capacity, and Table I gives the relative weight, cubical space and floor space.

TABLE I.

	Weight.	*Cubical space.*	*Floor space.*
Twin-beam air pump	14,500 lbs.	375 cu. ft.	33 sq. ft.
Two air ejectors and one condensate pump..	2,100 lbs.	18 cu. ft.	8 sq. ft.

Other comparative advantages of the air ejector are:—

It has no moving parts nor valves, does not require lubrication, nor attendance during operation.

Its operation is noiseless, simple and reliable.

No foundations are required and no restriction as to location exists. It is to all intents and purposes a piece of pipe.

Since there are no moving parts there is no wear, and consequently the maintenance costs are practically nil.

When using a steam-air ejector the feed water carries less absorbed air into the boilers than when using a twin-beam pump. The condensate pump forms a self-contained unit. All the attention it requires is to see that its ring oiling bearings are kept filled with oil. The discharge head of the condensate pump is not limited, as is the case with the twin-beam air pump. Since it operates independently from the air ejector the pump has greater flexibility and can easily take care of large amounts of condensate without influencing the air-handling capacity. This is not possible with the twin-beam air pump, in which the air-handling capacity is limited by the piston speed and simultaneously by the amount of condensate removed.

When using air ejectors and condensate pumps a considerable saving in weight and cost of piping can be made because the pipe sizes are smaller, as much higher velocities are permissible when dealing with water and air separately.

Steam-air ejectors are generally supplied in pairs for marine condenser installations, this subdivision having the advantage of reducing the steam consumption by operating with one under light loads or when the installation is air-tight, also of providing additional capacity in case of unexpected leakage. In larger installations sometimes three or even four air ejectors working in parallel are being used with two condensate pumps.

This subdivision also gives an absolute assurance against a total shut-down, which may easily occur when using a single reciprocating type air pump.

As the condensate pump forms an essential part of an air-ejector installation a short description of several types of condensate pumps will be given.

A centrifugal type of condensate pump, driven directly by a steam turbine, which has been extensively used, is shown in Plate 78. It has a single-inlet impeller mounted on the overhung end of the turbine shaft. The pump casing is bolted directly to the turbine casing, thus forming a rugged unit with only two bearings. This construction makes it impossible to throw the shaft out of alignment when bolting the pump down to the foundation or when connecting the piping. A vent is provided ahead of the inlet of the impeller and connected to the condenser to permit any air which may have been drawn into the pump through whirling of the condensate as it enters the suction pipe to return to the condenser and prevent the pump from becoming air-bound. The turbine is running at a constant speed which is regulated by a speed governor; it also has an emergency trip. Carbon packings are generally used on the turbine glands. The pump has to remove the condensate under a high vacuum, and therefore has to overcome a suction head of 30 feet or more. This makes it necessary for the condensate to flow by gravity to the pump impeller, and it is advisable that the pump be submerged not less than 2 feet below the lowest water level of the condenser.

Direct-acting reciprocating piston pumps, either vertical or horizontal, are also used for the removal of the condensate. Their design is adapted to the features peculiar to condensate pump service. Their submergency is more than that for the centrifugal type and is never less than 4 feet from the top of the discharge valves to the water level of the hot well. The stuffing boxes are water-sealed, and the piston speed does not exceed 25 to 30 feet per minute when handling the normal capacity. The speed of the pump is controlled by a float arranged in the hot well of the condenser so as to prevent racing of the pump at light loads.

The arrangement of an air-ejector installation with turbine-driven condensate pump is illustrated in Plate 79. Live steam at boiler pressure is delivered through the steam strainer to the pressure regulator; passing through the pressure regulator the steam is delivered to the main steam valve of the air ejectors at practically a constant pressure for which these have been designed to operate, the pressure regulator taking care of variation in the boiler pressure. The suction of the condenser is connected to the air suction of the ejectors by a tee, a gate valve being arranged so to permit the shut-off of each ejector. The discharge of the ejectors leads into the feed and filter tank.

A swing check valve is arranged close to the tank to prevent any water from rushing back into the air ejector if shut down before breaking the vacuum in the condenser.

Inside the feed and filter tank the mixture of steam and air is discharged through a submerged pipe perforated at the bottom. By coming into contact with the water the steam is condensed while the air, forced to rise by the upward flow of the water, is liberated, escaping through a vent arranged in the top of the tank. Water is continuously supplied to the feed and filter tank by the turbine-driven centrifugal condensate pump, which delivers the condensate into the filter tank through a swing check valve, gate valve and a perforated pipe with holes diametrically opposite to that of the ejector discharge pipe. A vent pipe is arranged between the suction inlet of the condensate pump and the condenser.

A re-circulating pipe, with a strainer, a thermostatically operated valve and a globe valve, is provided between the feed and filter tank and the condenser to insure at all times a sufficient water supply for condensing the steam from the air ejectors.

During maneuvering or while at sea, the main engine or turbine supplying the condenser with exhaust steam may stop, consequently very little condensate will be delivered by the condensate pump to the feed and filter tank, and not sufficient to condense the discharge from the air ejector which at the time is in continuous operation.

To prevent a rise in temperature of the water in the feed tank over that desirable (usually 140° F., depending, however, on the general arrangement of each individual installation), a thermostat is arranged in the feed and filter tank, controlling the thermostatically controlled valve. As soon as the water reaches the temperature for which the thermostat is set, the latter will open the thermostatically controlled valve and the vacuum will draw water through the re-circulating pipe into the condenser. By falling over the cool condenser tubes (the circulating pump, of course, being in operation), this water will be cooled and returned by the condensate pump to the feed and filter tank, thus preventing any further rise in temperature until the exhaust from the main engine again supplies sufficient condensate.

This arrangement makes the air ejector installation absolutely reliable, even if operated by inexperienced attendants. Starting is easy, as it requires only the opening of two steam valves, one for the turbine driving the condensate pump and one for the air ejector. During operation no attention is required, and, to shut down, only these two valves have to be closed.

A diagrammatic arrangement of an air-ejector installation with a direct-acting reciprocating piston pump is shown in Plate 80. This arrangement is generally applied to marine reciprocating engine surface condensers, and is similar to that for marine turbine surface condensers already described. The only difference is that the speed of the reciprocating piston pump is controlled by a float arranged in the hot well.

The steam-air ejector was first tried out on board a ship about four years ago. It was feared that, by discharging the co-mingled steam and air into the feed tank, the amount of air absorbed by the feed water would be greater than that when using a reciprocating piston air pump. Comparative tests have shown that this fear was groundless. The amount of air absorbed in the feed water, when using the ejector, was .35 per cent less than that when using the twin-beam air pump.

With this obstacle removed, the air ejector rapidly gained the confidence of shipbuilders and marine engineers, so that there are to-day to the writer's knowledge over five hundred ships built in this country equipped with steam-air ejectors. About one-third of these have been in successful operation for periods up to two years.

From this it would appear that the steam-air ejector to-day occupies a position among the air pumps similar to that occupied by the steam turbine among steam-operated prime movers. Its advantages, reliability and simplicity are dominating factors which will gradually make its adoption as universal as that of the steam turbine.

DISCUSSION.

The President:—Mr. Kothny's paper, "New Developments in High Vacuum Apparatus," is now open for discussion.

Mr. Martin L. Katzenstein, *Member:*—I have been unable, since the receipt of this paper, to prepare a complete reply, and the limited time for discussion would hardly permit such a presentation, even if considered desirable. In many ways the paper is interesting, and I agree with certain features in the discussion of the operation of the ejector itself. In other respects the paper, while interesting, is misleading and quite incorrect. In order to save the time of the members, I have outlined briefly a discussion of the principal points to which I would like to call attention.

The development of the marine condensing unit has not kept pace with stationary practice. A vacuum corresponding to an absolute pressure of .5 to .6 pound is maintained in many of the more important public utilities installations during the winter months, and the vacuum producing apparatus has reached the highest stage of efficiency. With full appreciation of the modifying conditions, I see no reason why, as stated on page 175, the limit for marine condenser performance should be placed at 1½ inches for 60 degrees or 2 inches for 70 degrees, whereas a stationary plant is designed for and readily maintains 1¾ and in some cases 1½ inches with 70 degrees, the latter corresponding to a saving of 2½ to 3 per cent in the steam consumption of the main turbine.

On page 176 reference is made to air leakage, and Plate 68 gives curves stated to show "permissible air leakage for marine surface condensers." Why the word "permissible" is used, I do not understand. The condenser must handle all the air, including leakage coming through from the turbine. Possibly the curve limits the air capacity of the ejectors as figured by the author, or it may have reference to the condenser efficiency which is definitely known to be reduced upon a straight line curve in direct ratio to the amount of air leakage. For discussion of this feature, I would refer you to the paper presented in May, 1916, at the Chicago meeting of the National Electric Light Association.

The curves upon Plate 68 would be very interesting if correct and authoritative. If they were determined by experiment, it would be interesting to see the determining data and observations; as a matter of fact, within the week I have had estimates given me by an engineer whose experience and opportunities for obtaining such data were at least as good as that of any engineer in the country. On the smaller sizes his allowances run from 50 to 100 per cent greater than shown on Plate 68. Mr. R. J. Kaula presented to one of the British societies a curve of leakage as reproduced in *London Engineering* of April 18, 1919, in which, for 50,000 pounds, he shows 41 pounds, and for 100,000 pounds of steam, 68 pounds of air per hour, as against 20 pounds and 32.5 pounds respectively as shown on Plate 68.

It will be interesting to state in this connection that the 30,000-kilowatt units installed at the water side station of the New York Edison Company (approximately 350,000 pounds of steam) are readily maintained in service at between 16 and 23 pounds of air leakage per hour, an amount which corresponds upon Plate 68 to a steam consumption of from 32,000 to 62,000 pounds per hour.

The New York Edison Company measures once a week, through a gasometer, the non-condensible vapors discharged from each condensing unit; when the leakage increases to, say, twice the above amount, the vacuum is so much affected that the unit is overhauled, at which time the leakage is brought as low as 13 to 15 pounds of air per hour, or at special times even lower. Other large stations similarly equipped report practically the same result. A complete discussion of this subject would make a lengthy paper, but evidently the curves on Plate 68 are hardly capable of miscellaneous application.

The separate wet and dry system referred to by the author has been used by Worthington for many years in stationary practice and has been extended to include vertical marine pumps of the twin type. The steam ejector has the important advantage for marine work of reduced space and weight, but many disadvantages and complications, as may be seen from the description and layouts accompanying the paper. The Worthington Company have carried on investigations and experiments in the ejector field for many years and are now offering ejectors either of the condensing or non-condensing type. The paper refers especially to the adoption of the air ejector during recent years and conveys the impression that this has been to the exclusion of the twin or twinplex type. The adoption of the ejector would have been a slow process except for the unusual conditions brought about by the war. The battleships, cruisers, and destroyers of the United States Navy were equipped with the twinplex type of pump (upwards of 400 during this period for destroyers alone), and in a few cases, where the ejector was installed by the Navy, it was only as an auxiliary or in conjunction with a pump of the twinplex type.

The Worthington Company's capacity and production were greatly increased during the war to care for the government requirements, and it was impossible to produce the additional vacuum pumps which would have been required by the cargo vessels, and equally impossible under those conditions to undertake the further development and manufacture of an additional line, such as ejectors.

The reasons for the building of the cargo steamers and the essential features of design were quite different from those either in the ordinary merchant or naval service, and a simple, light, cheap apparatus that could be quickly produced and installed was therefore permissible.

This is no place to introduce a discussion of the Worthington ejector, but I will state that the results obtained, both in economy and elasticity, have been in some respects quite unexpected and exceed those reported for other types of ejectors.

Plate 69, as referred to on page 176, shows the well-known curve illustrating Dalton's law, but these curves are very misleading in their application to the steam ejector. The curves give the weight of air and steam present in a mixture, but say nothing with regard to the specific gravity and volume, which are the important factors in the capacity of the steam ejector. The total weight, as obtained from these curves, will give the actual work performed and will show, in the case of any steam ejector, a ridiculously low mechanical efficiency, but it does not measure the capacity or steam consumption of the ejector, which is absolutely a volumetric machine. For example, at 90 degrees and 2 inches, the curve shows 1.57 pounds of water per pound of air, or a total of 2.57 pounds; at 80 degrees saturation, 0.67 pound of water per pound of air, or a total of 1.67 pounds; at 70 degrees 37 pounds of water per pound of air, or a total of 1.37 pounds; so that at 90 degrees the total weight per pound of air is 54 per cent greater than at 80 degrees, and 87½ per cent greater than at 70 degrees, but the ratio of the volumes is entirely different. Those depend upon the partial air pressure; at 90 degrees the volume is 61½ per cent greater than at 80 degrees, and 116 per cent greater than at 70 degrees, so that, in handling questions of capacity and steam consumption of the ejector, it is the relative volume and not weight that must be considered. The paper in an indirect way refers to this at the bottom of page 180, where the statement is made that careful experiments have shown the amount of air handled actually less than the calculated theoretical volume based upon weights alone.

I will not offer any mathematical discussion of the ejector or the varying conditions in the different stages, but would call attention to the likelihood of confusion in interpreting Plate 71. This would form a very unfair basis of comparison with any other type of pump: for instance, an hydraulic vacuum pump to rotative dry vacuum pump. Also, while a dry air orifice test in the shop is practically the only means of obtaining the capacity of the steam ejector, it must be considerably corrected before it can be applied to actual service. If in service we removed the mixed vapors from a condenser at 2 inches absolute pressure (corresponding to 101.25 degrees), and the vapors are cooled in leaving the condenser, say, approximately 10 degrees or 91.83 degrees, this will mean, according to Dalton's law, that the steam pressure will be 1½ inches, but the partial air pressure only ½ inch, and the partial air pressure determines the volume of the vapors to be handled by the ejector. In the shop test, with dry air only at 2 inches total pressure, the ejectors will then evidently handle four times the air that it will in service. In other words, we should divide the quantity of air obtained in the shop by four to obtain the amount of air that will be handled in actual service as it comes saturated from the condenser. The actual amount of cooling of non-condensible vapors will vary with the quantities of steam, air and water as well as the temperature of the water.

Worthington has found a decided advantage in having a slight degree of superheat in the steam operating the ejector.

It is unfortunate the author did not give actual figures or calculations in connection with any of the statements made, so there is no opportunity for checking up. The statement, however, is made that a steam ejector with a turbine-driven condensate pump will require less steam than a twin beam air pump or even a twinplex pump with the wet and dry cylinder. This is absolutely incorrect, and from actual tests we find that the twinplex pump will take only about 40 to 45 per cent of the steam required by the ejector and condensate pump, operating at the same capacities. There is considerable misinformation in various circles with reference to the twin pump. There have been papers extensively circulated abroad, comparing twin pumps unfavorably with other types of vacuum apparatus, but, where definite observations were given, they are ridiculous, in comparison with the known performances as readily tested out. For one thing, dry air was used in their shop test, the air being drawn over water at a given temperature, and the assumption made that the air became 100 per cent saturated at the temperature of the water, which is incorrect, as is perfectly apparent. Any pump must reduce in efficiency as the temperature of saturation nears the temperature corresponding to the total pressure, and in the papers referred to a zero efficiency is given at the point where an actual efficiency of between 30 and 35 per cent is readily maintained.

Another important point is that the builder is seldom asked to specify the size or guarantee the performance of a twin pump. The size has been specified and definitely asked for and is ordinarily much larger than necessary. Doubtless those of you who are interested have noticed, in the various tests of the vessels of the U. S. Navy as reported through the *Journal of the Society of Naval Engineers,* that the speed of the twin pump is seldom more than one-third to one-half of its normal rated speed, and in many of those trials where the manufacturer's representative has desired to reduce the speed of the pump very materially, in order to obtain a better economy of the pump, he has been instructed to keep the speed up in order to obtain the amount of exhaust steam necessary to heat the feed to the proper temperature.

The efficiency of the twinplex pump is much greater than that of the twin pump, and it covers the requirements of the separate wet and dry vacuum system.

The steam ejector has not been adopted in the high grade stationary plants for various reasons, one of the principal being the fact that perhaps the greatest economical development of turbine-condensing plants in recent years has been due to the adoption of the wet and dry system alluded to by the author. In that system the non-condensible vapors are handled by a separate pump and are thrown away while the feed is pumped into a closed heater (or open heater with cover) and recirculated without this air; consequently, in a closed system, a very small amount of air is present. Tests reported some years since (American Society of Mechanical Engineers, Transactions 1912, page 737) upon a plant at the Boston Elevated Station showed that a dry vacuum pump, which would normally operate at from 100 to 125 revolutions, showed no drop in vacuum if operated at around 45 to 50, which was as slow as this pump would operate without centering. The speed was reduced by hand control to 8 or 9 revolutions per minute without drop in vacuum, and the pump was finally closed down entirely; in one-half hour's operation of the turbine with the vacuum pump stopped, the drop in vacuum was only 0.3 inch.

The use of the steam ejector can only be countenanced where the heat in the steam can be reclaimed for heating the feed, but in order to do this the steam must be discharged through the feed water, which is thus fully recharged with air. I have alluded earlier to the effect of air in reducing the efficiency of the condenser.

There is another most important feature already referred to in connection with Plate 69. When the non-condensible vapors leave the condenser to enter the steam ejector, they are seldom cooled more than 10—perhaps 15—degrees below the temperature corresponding to the vacuum. But where an hydraulic vacuum pump is used (which uses hurling water of the temperature corresponding to the circulating water), the vapors are reduced to or saturated at the temperature of the circulating water. The same is also true with respect to the rotative dry vacuum pump, in which the vapors are saturated at the temperature of the jacket water (which is the same as the entering circulating water). This important characteristic of the rotative dry vacuum pump, as well as the gasometer system referred to as existing in the New York Edison Company, is described in a paper presented in 1913 to the National Edison Companies.

Where a steam ejector carrying 2 inches absolute in service received the vapors cooled to, say, 90 degrees, the hydraulic vacuum or rotative dry vacuum pump would handle vapors cooled to or saturated at 70 degrees which would occupy approximately only 46 per cent of the volume of the vapors going to the steam ejector, so that the steam ejector does 2.16 times as much work as it would if its vapors were cooled to the same temperature.

Nothing has been stated in this paper with regard to the mechanical efficiency of the ejector, but, before closing, I would like to call attention to the fact that at a recent convention superiority of the steam ejector was claimed because all the heat is saved, including that equivalent to the work performed. In reply figures were submitted showing that the amount of useful work performed by the steam ejector represented in British thermal units only .003 of 1 per cent of the total evaporative heat of the steam delivered to the main turbine. This also gives some index of the efficiency of the steam ejector.

The author states that the steam required by the non-condensing ejector will be reduced 30 to 40 per cent by the condensing ejector. The Worthington ejector tests show a reduction of approximately 50 per cent; had the author given actual steam consumptions, then a study of the heat balance could be made. It is not necessary to pump the circulating water through an inner condenser and thus waste the heat. On the contrary, the condensate is pumped through this condenser on the way to the open (or closed) heater, thus saving this heat, and recent installations have been called for where an after-surface condenser as well as inner-condenser is required in order to save recharging to the feed with air which occurs when the ejector discharges into an open heater. The condenser offered for this service combined both the inter-condenser and after-condenser in one shell.

It will be found that the percentage of steam saved by using the condensing type of ejector is a very large factor and not to be argued down without a careful study of the heat balance. It should be remembered, however, that the heat balance on a ship is very different from that in stationary plants, where frequently the auxiliary exhaust steam to be utilized must be limited to an amount which will not raise the feed to more than 120 or 130° F., the feed being further raised by being passed through economizers and other heat-reclaiming devices. The statement is made that the amount of air absorbed by the feed water, when the ejector is used, is 35 per cent of that when the twin-beam air pump is used. The twin-beam air pump has been practically superseded in the latest installations by the twinplex or wet and dry pump. It is not believed that actual tests will show any increases over the ejector in the air absorption by the feed. This, however, would be to some extent modified by the design of heater and whether the exhaust passed into the body of the feed or through a falling spray.

The layouts accompanying the paper show a very complicated system with special regulating devices and elaborated heater and filter tank which must all be taken into account in connection with both the weight and space occupied, and where the heat balance requires a condensing ejector, both the weight and space are also effected and the twinplex system will be found much the simplest and most advisable.

With regard to the weights figuring upon the same quantity of air in both cases, the twinplex runs from 2 to 2½ times greater weight than the non-condensing ejectors with turbine-driven condensate pumps, not approximately 7 to 1 as stated in the paper, which evidently is comparing a very large capacity old-type twin with an ejector of limited capacity. In the case of many cargo vessels, the items of space and weight are not of prime importance.

It must be remembered also that the steam ejectors, on account of lack of elasticity, must be installed in multiple units, and I would like to specially impress upon the members the fact that the vacuum pump handles air and not steam, and the old basis of attempting to rate pumps upon the quantity of steam handled must, before long, give way to an intelligent selection of size based upon the quantity of air used. Referring again to Plate 68, assume that the quantity of air is correctly given for 30,000 pounds steam as 15 pounds per hour and for 90,000 pounds as 30 pounds per hour; the fact is that if the latter turbine, having a normal steam consumption of 90,000 pounds, is operated at partial load, requiring only 30,000 pounds of steam, the air leakage will remain 30 pounds as given for the 90,000-pound condition, provided it is not greatly increased owing to additional leakage at partial loads as is the case in many designs of turbines.

The President:—Is there any other gentleman who wishes to offer any comments on this paper?

Mr. William W. Smith, *Member:*—Mr. Kothny has contributed a very valuable paper. It is original, covers the subject thoroughly, and contains much information which is useful in condenser design. It is thought that more attention should be paid to the scientific design of condensing plants, especially so in the case of cargo vessels. There are several elements that require to be considered in this connection, such as the velocity of the circulating water in the tubes, the surface used, the quantity of cooling water, the temperatures prevailing and the heat units to be transferred.

In this connection, the table given by Mr. Kothny on page 175 is interesting, because he states the different vacua which you might say are good practice, and which are recommended for different temperatures of circulating water. It is quite important to note that the vacuum is principally dependent on the temperature of the cooling water, and that extremely high vacua cannot be obtained with warm cooling water such as prevails under tropical conditions.

In this paper Mr. Kothny gives a vacuum of 2 inches, absolute for sea water of 70° F. That appears to be the universal practice, and is considered a good rating for a marine plant. If this design condition prevails—that is, 70° F.—then you will get different vacua at the other temperatures of sea water. Very often the owners, in specifying the condensing plant, will call for a vacuum of 28.5 inches at 85 degrees air temperature, which, of course, is quite impracticable.

Mr. Katzenstein in his discussion brought up the point of maintaining the vacuum at the absolute pressure of a half inch of mercury for marine turbines. Turbines are not usually designed to utilize as high degree of vacuum, and it would not be of much value if it were produced. Most turbines will expand the steam to about 28.5 or 28.75 inches at full power.

It should also be noted in this table that 60 degrees is the lowest temperature of sea water given. Of course the temperature actually encountered will go much below this, and sometimes will go to 30 degrees, in which case a higher vacuum would be obtained. In many cases the operating force on board ship will run the circulating pumps at high speed with a cold sea-water temperature and maintain a vacuum well above 29 inches, when actually they are losing steam by doing so, because the turbines are not able to utilize efficiently that high degree of vacuum.

The table on page 183, giving a comparison between the twin-beam air pump and two air ejectors and one condensate pump, is substantially what we found in an investigation for our vessels of this subject, and, as will be noted, there is quite a difference in the weight, and in the space also. The weight, the space and the arrangement, of course, are matters of study for each particular installation. In the case of the vessels constructed by the Federal Shipbuilding Company, we found there was a material advantage, taking everything into consideration, for air ejectors, and for that reason we installed air ejectors on all the thirty vessels which we constructed for the Emergency Fleet Corporation, in addition to a number of other vessels which we now have under construction.

On page 184, the author refers to the use of piston type pumps for the removal of condensate. I call attention to this because of the idea that a good many people have that only a turbine-driven centrifugal pump can be used for the removal of condensate. We have prepared designs for the use of both turbine-driven condensate pumps and piston-driven condensate pumps. We selected the turbine pump, since to us it appeared to have some advantage —at least for turbine-driven ships, where the men are familiar with that type of machinery. The piston condensate pump, of course, can be either of the vertical or horizontal type, depending on the particular arrangement of the installation where it is used.

The piston speed of the condensate pump as given by Mr. Kothny appears to be a little low. Of course this is somewhat a matter of choice, and naturally it allows for a larger factor of safety in the capacity of the pump.

On page 185, the author refers to a re-circulating pipe, with a strainer, thermostatically operated, etc. It is not necessary to provide automatic circulation of the water for preventing high temperatures in a feed tank. This can easily be handled by hand, and, in the later vessels which we have built, we have eliminated the thermostatic control because we found that it worked quite well enough in this way. The thermostat, however, is a refinement which, if kept in proper order, will undoubtedly add to the luxury of the installation.

In regard to Mr. Katzenstein's comments on Plate 68, I hardly think these criticisms are constructive. It is evidently very difficult to state what the air leakage into a condensing plant is going to be, since it will vary so much with the design and construction of the installation. Yet the designer must have some leakage to use in the design of his plant. His assumed leakage may not be exactly right, but he must have something that he can use for determining the size of the air pump which is to be used. In fixing the capacity of most pumps, certain standard conditions of construction and installation of the plant are assumed, together with a standard air leakage for this design and type of plant. A pump is then provided which will take care of this air leakage and a great deal more; that is to say, a factor of safety will be provided for handling this quantity of air. The quantity of air to be handled in any given installation is an arbitrary value. It depends on assumed standard conditions, and I think that Mr. Kothny's effort to reduce it to some sort of a standard is of value. It is certainly better than no standard at all. As far as I have been able to find out, this is the way it is done in most cases. If the air pump is anywhere near large enough (that is, if it is based on good practice) and if you do not get the required vacuum, you go around and look for the leaks and make them tight until you get the required vacuum. We cannot say exactly what the air leakage is going to be, and must provide a reasonable excess of capacity. It is thought marine engineers should get together and decide on what are reasonable air leakages for different types of plants. I think, however, that Mr. Kothny's curves are useful and believe they are safe for designers to use.

The illustration on Plate 77 shows a comparison between the piston type pump and the air ejector. I have not checked this comparison, but, offhand, it looks about the same as we laid out in our ships.

In regard to the complexity of the installation, we have not found it specially complex. The feed tank which was mentioned as being somewhat special is really not special at all. It merely has two pipes projected into it, with perforations. That is the only special feature about it, except the re-circulating pipe which is connected on to it to carry the water back to the condenser for cooling.

Attention is called to Plate 80, showing an air ejector installation for a steam-engine vessel. I do not believe it is generally understood that air ejectors can be used to advantage for steam-engines. I have not made an exact comparison of the two types for engine vessels, but I would say, offhand, that there was not any disadvantage in using air ejectors. For most cargo steamers I think it is desirable to leave off the automatic thermostatic control, although that is something that does not do any harm, and may do some good.

We have turned out twenty-seven vessels with the C. H. Wheeler "Radojet" installation, and in all cases these vessels have given excellent results. The condensing plant was carefully designed all the way through. The specifications call for a 28-inch vacuum with a sea-water temperature of 70 degrees, and in practically all cases, after the leaks have been taken care of properly, we have obtained from ¼ to ½ inch better vacuum in service than we designed the installation for, which is considered a good performance.

In regard to the steam consumption of the piston pump, and the air ejector, it would be interesting to have accurate figures on the two steam consumptions. We made a comparison of these values in selecting the air pumps for our vessels, and, as I recall it, the Radojet was less than the piston pump, although I would not be positive. We found that the Radojet was much lighter, and we considered it simpler. It was more compact, and we considered that it gave us a better engine-room arrangement—more floor space and more room around the engine.

There is one feature in connection with the use of these air ejectors for turbine-driven vessels which should be taken care of, and that is the turbine drainage. The turbine drainage is of equal importance with the lubrication for the safe operation for turbines. Great care must be taken to provide a safe drainage for the turbine, so that the water will not back up and interfere with the blades and cause stripping. In some of the vessels which we built we installed steam ejectors; in some small piston pumps on the tank top, and in others the two together. We installed both an ejector and a steam pump and provided suitable means for observing the water in a drain well, so that the operator could keep track of it. I want to emphasize the importance of providing for absolutely safe drainage where these air ejectors are used.

In regard to the heat balance and auxiliary exhaust steam, etc., in some cases you can control this in your design and in other cases it gets away from you. In most cases, in ordinary cargo ships the auxiliary exhaust is all condensed in the feed heater. In most turbine vessels you have more auxiliary exhaust than the feed heater will take care of. The surplus is carried into the low-pressure turbine and does useful work. The exhaust steam develops a horse-power with about double the consumption of high-pressure steam, which is not a bad performance. In other words, in the complete expansion turbine, with high-pressure steam, we get a horse-power for about 11.5 pounds. With the exhaust steam we develop a horse-power for about, say, 25 pounds, so that the exhaust steam going into the low-pressure turbine develops quite a good deal of power.

The steam-air ejector is a new device to most operating engineers, and for that reason we were very careful to provide the operating engineers of our vessels with suitable information as to the practical operation and handling of these air ejectors. We prepared a complete set of practical operating instructions, which were as practical as we could make them, and furnished them to the operating engineers of our vessels. This was in addition to the instructions which were prepared by the makers of the apparatus, and I think that the supplying of this information had a good deal to do with the success which this apparatus has met, because we have had absolutely no trouble whatever with any single ship we turned out with this apparatus.

It is very important also to provide instructions for the draughtsmen and designers in handling a new type of apparatus of this character, and we provided instructions for our drawing office which took care of the various details in a proper way.

In closing, it might be interesting for Mr. Kothny to state the number of vessels which have been fitted with air ejectors of this and other types, if he has that information; also the types of installations, whether engines or turbines, and the total horse-power of steam vessels which have been equipped with air ejectors.

Mr. Ernest H. B. Anderson, *Member:*—There are one or two points I should like to draw attention to. It is rather surprising that the author did not draw attention to the work of Sir Charles A. Parsons, who invented the vacuum augmentor in the year 1903, for both the Radojet and the Le Blanc air ejector are based on the same principle.

The Parsons vacuum augmentor is in use in practically all the large turbine-driven battleships of the American Navy, and in most of the destroyers.

The great feature about the vacuum augmentor lies in the fact that it is used in conjunction with the ordinary single-acting wet-air pump, and the air pump is so arranged in the engine-room that it can be used with or without the augmentor.

In all turbine installations it is of vital importance that the turbines are at all times drained thoroughly and kept clear of water, but I note that the author does not take up this question, although it bears a very essential relation to this apparatus.

Experience has shown that the turbo-condensate pumps do not operate properly if vapor instead of water passes into the suction pipe, and for this reason it is necessary, in cases where this form of pump is fitted, to install an ejector to deal with the turbine drains. This is not a desirable feature, and experience has shown it is necessary that the drainage from the low-pressure turbine exhaust will be entirely automatic.

It seems to me that the use of high-speed turbo-condensate pumps will soon be superseded by the regular twin-beam, single-acting, wet-air pump, largely for the reason that this type of pump can be subjected to all kinds of rough usage without showing signs of distress.

The President:—Are there any further comments? If not, we will ask Mr. Kothny to make such rejoinder as he desires.

Mr. Kothny:—I cannot but feel gratified to see that this paper has elicited such an interesting discussion. The subject is obviously one which interests marine engineers, and one of the reasons for submitting this paper to this Society is that I have been personally associated for the last ten years with the development previously mentioned. It is undoubtedly of further interest, in view of the revolutionary nature of the development.

When I prepared this paper I had in mind just to record this development, but not to talk about the advantages of the different makes of air ejectors and the different types of air pumps.

Mr. Katzenstein, in his discussion, first criticised Plate 68, showing the amount of permissible air leakage. He quoted some figures from a British paper, which showed 41 pounds of air leakage per hour for 50,000 pounds of steam. The curve illustrated on Plate 68 shows only 20 pounds per hour air leakage for 50,000 pounds of steam. Immediately afterwards he stated that, on a turbine consuming 350,000 pounds of steam per hour, the maximum permissible air leakage was only 16 to 23 pounds; in other words, considerably less than the amount taken from the curve on Plate 68.

From this it is not quite clear what Mr. Katzenstein wishes to convey—whether the limits of air leaks given on Plate 68 are too small or too large. Mr. Smith correctly stated that the air leakage is an arbitrary value, and you will agree that there should be certain limits for this arbitrary value. The curves published in Plate 68 should not be taken as absolute in every sense—they were published more with the intention of suggesting a standard, so that marine engineers interested along this line could get together and establish a standard limit for this arbitrary value, as has been done in many other branches of engineering.

Mr. Smith also stated that on the ships on which they used the steam-air ejector the vacua obtained were generally from 0.35 to 0.40 of an inch higher than those guaranteed. This is interesting, and I am glad to hear it because the sizes of the air pumps selected for these installations were based on the air leaks shown in Plate 68. This would also indicate that the values given in the curve on Plate 68 are amply large, and not too small, as Mr. Katzenstein stated.

Mr. Katzenstein also criticised Plate 69, showing the relation between vapor and air. I do not think I need comment on this, because this curve is based on physical laws, and I do not pretend to be author of the curve—it is a curve which may be found in many text-books and which have been previously published, and I have no doubt the curve is correct.

Mr. Katzenstein called attention to the likelihood of confusion in interpreting Plate 71, illustrating the characteristics of an air ejector, and said that it would form a very unfair basis for comparison with any other type of pump. However, these statements are at variance with his explanations which followed them. On page 177 of the paper it is clearly stated that the curves illustrated on Plate 68 only give the amount of free air entering by leaks, but not the air-handling capacity of the pump. The latter has to be calculated by using Plates 68 and 69. Consideration should be also given to the fact that the work of compression of a mixture consisting of air and vapors is somewhat different from that of dry air, This, however, applies to any type of dry-air pump, whether air ejector or rotative dry vacuum pump.

Mr. Katzenstein's statement that the twin-beam air pump will take only 40 to 45 per cent of the steam required by the ejector is incorrect. The ejector plus condensate pump takes less steam than the twinbeam or twinplex air pump, when handling the same amount of air and condensate. Comparative tests based on the same air and condensate handling capacity, under identical operating conditions, have given the following results: Twin-beam pump with cooler, 1,465 pounds; air ejector, 925 pounds.

In regard to his remark that twin-beam air pumps generally are operated at from one-third to one-half of their normal speed, it would have been very interesting to have some shipbuilder or marine engineer comment on that point. My experience with twin-beam pumps or twinplex pumps has been somewhat limited, but reports from operating marine engineers do not coincide with Mr. Katzenstein's remark.

It is perhaps unnecessary to reply to the numerous remarks made by Mr. Katzenstein in regard to stationary plants, because this Society consists of marine engineers, and I do not think you are interested in proceedings which have no connection with marine engineering.

As to his statement that the rotative drive vacuum pump is in a position to handle vapor at a considerably lower temperature than an air ejector, I cannot agree with the same. If Mr. Katzenstein's assumption that the vapors are cooled inside the cylinder by the jacket water were true, the air pump will have to remove at each stroke a certain amount of water. Every one knows very well that the dry vacuum pumps have very close clearances, and if there is water in the cylinder, which is not compressible, something else will have to give way, viz., the cylinder head or the piston.

As far as the mechanical efficiency of the steam air ejector is concerned, it is known that this efficiency is low, as is the case with all dry-air pumps. But the mechanical efficiency is not of such great importance as the thermal efficiency, since the latter influences the fuel consumption. The thermal efficiency of the air ejector is as high as 98 per cent. All the heat which enters with the live steam is returned directly to the feed water, with only a very small amount of loss, consisting of loss through radiation in the steam pipes and the apparatus itself.

Referring to Mr. Katzenstein's remarks in regard to inter-stage and after-condensers, the use of the condensate for circulating water purposes necessitates the arrangement of a number of complicated automatic appliances, as operating conditions vary. It also requires a rather complicated piping arrangement. For these reasons inter-condenser air ejectors do not appeal to marine engineers, and in over 600 ships with air ejectors there is not one with inter-stage condensers.

The figures given in Table I refer to the up-to-date twinplex air pump and were obtained from the manufacturer of this pump. They are based on equal air and condensate handling capacity; that is to say, both pumps removing the same amount of condensate in pounds per hour, and also the same amount of air in pounds per hour, under equal operating conditions.

Mr. Katzenstein also said that weight and space requirements are not of prime importance in cargo vessels. I leave it to the marine engineers to pass judgment on the correctness of this statement. Mr. Smith has already confirmed the statement that weight and space requirements are of considerable importance.

Mr. Katzenstein also mentioned that in some marine installations air ejectors had been applied, but always in connection with twin-beam air pumps. It is true that there have been several installations where the air ejector was used in connection with a twin-beam air pump. The request for making those installations was based on the fact that the necessary vacuum could not be obtained with the twin-beam air pump, and the air ejector was installed to improve the vacuum. The twin-beam air pump was used only for taking out the condensate from the condenser, and in that case was run at about one-third of the normal speed.

Mr. Smith stated that the piston speed for the reciprocating type condensate pumps as given in the paper is rather low. That is true, but consideration must be given to the fact that these pumps generally are very small and have a short stroke; also that they should be capable of handling twice the normal capacity in case of emergency.

It is therefore advisable to be rather conservative regarding the piston speed. A low piston speed will also help to increase the life of the pump.

Mr. Smith requested me to state the number of vessels which have been equipped with steam air ejectors. There are today over six hundred ships which have been equipped with air ejectors in the United States. About half of this number have been in service for periods from one to three years. The total horse-power of marine condensers equipped with air ejectors is about 5,000,000.

Mr. Anderson called attention to the fact that Mr. Parsons' development in regard to the augmentor was not mentioned in this paper. Mr. Parsons certainly deserves all the credit due to him for the work which he has done in connection with the development of the vacuum augumentor. However, when I prepared this paper it was my intention to record some new developments, and it did not occur to me that I should speak about the Parsons augmentor, as this augmentor was brought out in 1903 or 1904.

Regarding the turbine drainage, I am glad that this point was mentioned by Mr. Smith as well as Mr. Anderson. It is very important that proper attention be given to this point, because, if the turbine is not drained, serious injury will result. The example stated by Mr. Anderson is a good one.

This matter is very interesting to the condenser builder, but he should not be blamed if improper provisions are not made. It is rather the duty of the man who makes the installation of the turbine to look after the proper drainage of the same. As Mr. Anderson stated, the condensate pump or air pump is not always located low enough to take care of these turbine drains, and in that case it is necessary to install either a drainage pump or small steam ejector for removing the condensate. Such installations have been made. I have seen a number, particularly steam ejectors, and as far as I know, they have worked satisfactorily. I have also seen some vacuum traps installed for lifting the drains from the lower part of the turbine cylinder into the feed and filter tank. Considerable trouble was experienced with these vacuum traps, because every time the trap operated a certain amount of air leakage would come in the low-pressure casing of the turbine, and the vacuum would drop. Most of these traps are designed for heating plants, where there is a vacuum of not more than 8 inches in the system, and they do not operate satisfactorily with high vacuum.

A steam ejector can be used very satisfactorily for removing the drains. The amount of steam consumed by the ejector is very small, if the ejector is properly installed. The ejector should discharge into the condenser and not into the feed and filter tank. All the work to be done by this small ejector is to raise the water about 6 or 8 feet from the bottom of the low-pressure turbine casing to the condenser. This can be done with very little expense of steam. I think that answers all the criticisms which have been made.

The President:—Of course the Society appreciates the paper presented by Mr. Kothny, and congratulates him on the extensive comment which has been made on it. The Chair therefore begs to present the Society's thanks to Mr. Kothny for the preparation and presentation of his paper. The time available for our morning session having expired, we will adjourn to reconvene at 2.15 o'clock this afternoon.

FOURTH SESSION.

FRIDAY AFTERNOON, NOVEMBER 14, 1919.

President Capps called the meeting to order at 2.20 o'clock.

THE PRESIDENT:—The secretary will read a list of applicants for membership recommended by the Council for election at a meeting of the Council held at the conclusion of the morning session.

SECRETARY COX:—These additional applications for the grade of Member were received and recommended by the Council for approval by the Society:

Members (27).

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THE PRESIDENT:—The recommendations of the Council for the election of certain applicants to the grade of Member are before you. What is your pleasure?

MR. ERNEST H. B. ANDERSON, *Member:*—I move that they be approved.

The motion was duly seconded, put to vote and carried.

THE PRESIDENT:—The first paper on the program for the afternoon session is entitled "The Launching of Large Vessels in Restricted Waters," by Captain H. M. Gleason, C. C., U. S. N., Member, and Lieut. Commander H. E. Saunders, C. C., U. S. N., Member. In the absence of the authors, Captain Geo. H. Rock has been good enough to consent to present the paper.

Captain Rock presented the paper.

LAUNCHING OF SHIPS IN RESTRICTED WATERS.

By Captain H. M. Gleason, Construction Corps, U. S. N., Member, and
Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

Although there have been ten papers read before this Society giving notes and data on the launching of various types of ships, the subject of the launching of ships in restricted waters has not been touched upon. The authors therefore have undertaken to present this subject based upon the experience of the Mare Island Navy Yard.

Unfortunately, there is very little definite information obtainable from text-books, technical papers, etc., giving the results of actual launchings in which means to check the speed of ships have been used. It is therefore believed that the subject-matter of this paper will be a welcome addition to the already published data on launchings.

The launching of a large ship is attended with a certain amount of risk under the most favorable conditions, and when there is added to this the problem of checking the ship after leaving the ways, the anxiety of those responsible is not relieved until the ship comes to rest. In most shipyards in this country there is sufficient water space in wake of the building slips to allow the ship free scope, or at least sufficient water space to check the ship by the dropping of anchors. In some shipyards situated on narrow waters the building slips are inclined at an angle of about 45 degrees to give greater travel.

Various methods have been successfully used to check vessels on leaving the ways, such as—

(*a*) The breaking of rope stops.
(*b*) The use of wood friction wedges.
(*c*) Fitting of a mask on the stern.
(*d*) Dropping of anchors.
(*e*) Slewing the stern with the channel by dropping stern anchors.
(*f*) Chain drags.

Anchors are generally fitted for emergency use in connection with any of the above methods.

The most commonly used method, especially in English and Scotch shipyards, is the use of heavy chain drags. The amount of chain used varies according to the experience at the various yards, and depends upon the nature of the surface available for the drags, launching speed, etc.; but the usual weight of chain is about onetwentieth of the launching weight to bring the vessel to rest in from 200 to 300 feet after leaving the ways.

The large building slip at the More Island Navy Yard is set nearly at right angles to the channel, which is 1,230 feet wide. It is therefore necessary to check any large vessel by other means than anchors, as the space available is not sufficient for direct checking or for slewing the stern. The Prometheus (fleet collier) was launched in 1908, using chain drags to stop her. Although the drags successfully stopped the vessel within 200 feet, the work necessary to pile the chain, to completely clear the slip of obstructions (blocking, shoring, etc.), and finally to untangle the masses of chain, was extensive and costly. The next vessel launched (August, 1912) was the fleet collier Jupiter, of 19,000 tons loaded displacement and 5,207 tons launching weight. The problem of stopping this vessel was gone into very carefully, and the final conclusion was to use chain drags. Friction brakes were considered, but the development of the idea at that time was not sufficient to warrant the trial. A description of the chain drags as used in launching the Jupiter will be of interest. In all 390 tons of chain were used, coiled in fourteen coils ranging from 10 to 50 tons each, the smaller coils or drags being arranged to take up first in succession to minimize the danger of parting the cables due to a too sudden stress. Seven piles of chain were placed on each side, connected up as shown below to three 2-inch (diameter) wire ropes attached to pads on the ship. The disposition of drags, wire ropes, and pads was as follows:—

Destination of Drags	Weight (tons)		Distance of stem beyond end of ways when drag takes up	Remarks
	Port	Starboard		
			Feet	
1-A	10	10	37	Attached to pad at frame 45
1-B	15	15	50	Attached to drag 1-A
1-C	25	25	60	Attached to drag 1-B
2-A	25	25	84	Attached to pad at frame 35
2-B	25	25	92	Attached to drag 2-A
3-A	45	45	124	Attached to pad at frame 25
3-B	50	50	307	Attached to drag 3-A
	195	195	Total, 390 tons.	

As a further precaution, two 3,000-pound anchors, one port and one starboard, were secured on the side of the ship at frame 136, each with 10-inch hawser stopped up at intervals and carried through stern chocks. One of these anchors was to be dropped on a signal given from the bridge should the drags fail to act. These anchors were intended to turn the stern up or down stream as seemed most expedient.

The actual results of the action of the chain drags were as follows:—

Velocity of ship at time of pivoting (also maximum) ....16.2 feet per sec.
Velocity of ship when fully afloat..............................15 feet per sec.
Distance run after first drag took up............................ 213 feet.
Drags 1-A—Port and starboard moved........................... 213 feet.
1-B—Port and starboard moved........................... 200 feet.
1-C—Port and starboard moved........................... 190 feet.
2-A—Port and starboard moved........................... 166 feet.
2-B—Port and starboard moved........................... 158 feet.
3-A—Port and starboard moved........................... 126 feet.
3-B—Were not moved.

The total weight of chain drags actually coming into play was therefore 290 tons or $\frac{\text{launching weight}}{18}$.

Early in 1913, in preparing for the launching of the next ship, the fleet oiler Kanawha, the development of friction brakes was actively taken up and experiments conducted which gave assurance of the practicability of friction brakes, using wire ropes passing through steel blocks under pressure. In the future design and development of these brakes advantage was taken of the data on this subject presented by Mr. A. Hiley, Associate Member of the Institute of Naval Architects, in a paper prepared by him and published in September, 1913, issue of *The Shipbuilder*.

The launching weight of the Kanawha was estimated at 4,000 tons (actually was 4,100 tons), and it was therefore estimated, from friction data obtained in experiments, that two launching brakes would be sufficient. A description of the launching brakes used and other precautions taken to check the ship is given as follows:—

"The brakes were securely anchored on each side of the slip, 22 feet from the center and about 80 feet aft from the bow. In each brake the friction length employed was two 2-inch diameter steel wire ropes (6 strands of 37 wires each), 600 feet long wound on reels fitted with brakes placed about 50 feet from the brake. Springs tightened by screws produced the requisite pressure upon the ropes, which are gripped between grooves in the upper and lower steel castings. In order that the pressure applied by the tightening screws might bear equally on the two ropes, the upper casting is formed with a ridge at the center to which the pressure is transmitted by beams formed of channel sections and plate. To regulate the pressure on the rope at will, three hand winches were provided and bull wheels fastened to the nuts over the spring washers. These were connected by a ⅜-inch (diameter) endless wire rope. The two friction cables were connected by a heart shackle to a single 2-inch (diameter) wire steel hawser, 560 feet long, which was shackled to a pad on the ship's side at frame 46 and on line with the second stringer, the same arrangement being used on both sides of the vessel. To assist in stopping the vessel a mask was also fitted on the stern, 16 feet wide by 11 feet high, the lower edge being 8 feet above the keel line. As a further precaution, a 11,000-pound anchor was secured on the port side of the ship at frame 112, with a 10-inch hawser stopped up at intervals and carried to the stern bitts. This anchor was to be dropped upon signal given from the forecastle should the brakes fail to act, and was intended to turn the stern down stream. A 6,000-pound anchor was housed in the port hawse pipe and was to be dropped if it became necessary to hold the bow."

The results of the launching, as far as they concern the launching brake, were as follows:—

Launching weight	4,100 tons.
Maximum velocity	16 feet per second.
Velocity when ship floats	13 feet per second.
Total distance friction ropes were drawn through brakes before ship was stopped	341 feet.

The same brakes and arrangements were used in launching the Maumee, a sister ship to the Kanawha. The results in the case of the Maumee were as follows:—

Launching weight	4,370 tons.
Maximum velocity	17.25 feet per sec.
Velocity when ship floats	14 feet per sec.
Distance cables were drawn through brakes before ship stopped	329 feet.

The same brakes, but without the stern mask, were used in launching the Cuyama, a sister ship to the Kanawha and Maumee. The results in the case of the Cuyama were as follows:—

Launching weight	4,056 tons.
Maximum velocity	16.7 feet per sec.
Velocity when ship floats	14.5 feet per sec.
Distance cables were drawn through brakes before ship stopped	444 feet.

By comparing the foregoing data the effect of the stern mask may be estimated as the Cuyama, without the mask and with a slightly greater velocity (when afloat) traveled 125 feet farther than the Maumee.

The action of the wire rope cables under pressure between the upper and lower cast steel brake blocks bears a very important part in the successful operation of this type of brake. There were no reliable data on the coefficient of friction under the actual working condition, and no guide as to what type of wire rope was best suited to obtain the desired frictional resistance. These two questions were solved by numerous experiments, and actual trials on vessels launched.

As to the coefficient of friction, the figure given by Hiley in the article previously mentioned is 0.08. All available information in handbooks and experiments on a small scale indicated, however, that the coefficient of friction was much higher than this, presumably about 0.2. To check this figure and to test the apparatus, the hydraulic brake developed for the California was mounted on a temporary stand and run with full pressure, using one friction wire in one of the grooves. The coefficient of friction deduced from these experiments is about 0.24, and the results obtained indicate that this value remains practically constant for all loads and all speeds of the wire rope.

As to the type of wire rope to be used, it was obvious that the wire strands on the outside of the rope would have to lie exactly parallel to the axis of the rope in order to prevent "rifling." It was evident, after the first experiments, that the wire would score the grooves in the cast-steel blocks, but so long as the scores remained parallel to the blocks, the action of the frictional pull on the wire ropes was entirely satisfactory, causing no unlaying or tightening up of the strands. There was some question as to whether a solid rope was necessary or whether a rope with hemp core would not be more suitable. The latter rope, being more elastic, is less likely to seize in the grooves, and for this reason has been used in all the launching operations with this brake. There had also to be considered the possibility of one or more strands breaking inside the brake and causing a jam. In this event the wire rope would likely be broken or the chain pendant to the ship carried away. During the several launchings, the outside wires in the strands have been perceptibly flattened by abrasion, but in no case has any strand parted or any wire rope jammed. In this connection it is interesting to note that the same wire ropes were used as friction ropes in the launching of three large ships, and now are in condition for use with at least as many more. From the experience gained at the Mare Island Yard the most satisfactory type of wire rope is 2-inch diameter, black, plow steel, 6 strand, 37 wires each, one hemp center, ordinary lay and 137 tons breaking strength. Particular care was required under the specifications for the rope to have the individual wires in each strand laid parallel to the axis of the rope at the point of frictional contact.

In connection with the launching of the battleship California the number of brakes necessary was determined from the data and experience in the use of the brakes used in launching the Kanawha, Maumee and Cuyama. In these latter cases two brakes were used with satisfactory results. Therefore, by comparison of the relative launching weights and velocities, the number of brakes required for the California was ten.

It was also considered necessary to have more definite control over the pressure to be applied to the friction blocks, and the screw, nut and bull wheel scheme was abandoned for hydraulic cylinders and pistons, as shown in photographs and plans. The experiments conducted showed that two pressure cylinders were sufficient in lieu of three.

Although previous experience with these brakes was sufficient to determine the number to be required in launching the California, the exact pull required from the brakes, collectively and individually, and the actual force required to stop the ship were not definitely known. There were also other questions on which more complete information than was available was considered necessary, such as the actual pivoting point at various tides, the depth of water required to accommodate the deepest dip of the stern, the behavior and clearance of the forefoot on leaving the end of the ways, etc. It was therefore decided to construct and try out a launching model based on the law of comparison similar to model tank experiments. This model could therefore be launched at will and as often as required, varying the conditions to suit those expected at the time of the launching.

The model was constructed with a length ratio of 1 : 96 (scale ⅛-inch equals 1 foot), as giving a craft which was easy to handle, yet sufficiently large to make possible a fair degree of accuracy in the results. Briefly, the dimensions of the model are as follows:

Length .... 6 feet 3 inches.
Beam ....12 inches.
Weight (approximate) ....39 pounds.
Material—Wood, hollow, finished in spar varnish.
Displacement, longitudinal position of center of gravity and longitudinal moment of inertia may be varied at will.

The general construction of the model is shown on Plate 83.

The tank and the framework supporting it, shown in the plates, require no special comment. Fresh water is used, and the contour of the river bed is represented by a layer of gravel and sand on the bottom of the tank. The water area represents the width of the channel, 1,230 feet, by a certain portion of its length, 320 feet. A modified form of hook gauge records the tide level in feet and tenths. The ground ways (with camber), the ship, the cofferdam bulkheads, crane piers, etc., are, of course, all reproduced exactly to scale.

To obtain correct results by the method of comparison, it is necessary for the model to run off the ways and through the water at its "corresponding speed." By running on a system of rails and steel wheels with hardened pivots it was possible for the model to accelerate itself at the required rate without the application of any external force. A central rail under the keel of the model runs on two large flanged wheels, one under the vessel and one at the end of the ways; two wheels at the fore poppets run on two rails which represent the ribbands of the ground ways. This system of mounting the model on three points was suggested by Mr. Percy A. Hillhouse and Mr. Wm. H. Riddlesworth in a paper presented by them at the Fifty-eighth Session of the Institution of Naval Architects, March 29, 1915, as being decidedly preferable to mounting on six wheels, especially when the ways were cambered. The forward keel wheel of the California model supports the weight until the keel track reaches the after keel wheel; the latter then supports the weight until the model pivots, when a special releasing device drops it clear of the forefoot as the model leaves the ways.

Although not shown on the photographs, the tumbling shores, cribs, wedges, cable reels and brakes were later added to the model, in order that launching drills might be held at the model, and the various gangs acquainted with their duties and the sequence of operations. A small brass preventer dog shore and a set of solenoid-operated mechanical triggers represent accurately the dog shores and hydraulic triggers on the ship. Two brass trimming masts, with pencils attached, erected at the forward and after perpendiculars, record the traces of the bow and the stern at all points of the launching operation.

A special recording mechanism was designed and built, to record simultaneously all data and to make the model as nearly as possible automatic in its operation. Reference to the plates will indicate the general arrangement of this machine. Without undue elaboration of details, the construction and operation of this mechanism may be described briefly as follows:—

A small cord (or cable, as it will hereafter be called) is fastened by a wire hook to an eye plate in the bow of the model at the 16-foot water line. This cable is led forward over an aluminum idler pulley carried on a swinging frame and then back and around a small drum about 1½ inches in diameter. Sufficient cable can be wound in a single layer on this drum to permit the model to run to the far end of the channel. The drum, as shown on Plate 87, has two silver contacts on a small commutator and acts therefore as a chronograph, giving two marks per revolution on the recording paper. A standard navy mean time break-circuit chronometer indicates seconds on the recording paper (by means of suitable solenoids and pencils) as a reference for the chronograph readings. The recording paper is that supplied for the Burroughs adding machine; it is drawn at constant speed over the paper table by two rubber rollers geared to a small D. C. motor. A small controller with adjustable segments, also driven from this motor, controls the current to the solenoid triggers and to the tripping coils of the brakes; this controller switches on and off the motor and other solenoid circuits and renders the mechanism entirely automatic in its operations. As the entire launch consumes only seven or eight seconds, it is not practicable to arrange for manual operation in this connection.

The launching brake mechanism, also to be very briefly described, does not, of course, operate in exactly the same manner as the brakes on the full-sized vessel. The small drum upon which the cable is wound is constructed with heads of highly polished steel. Cork insert brakes, carried on swinging plates, bear against the heads of the cable drum and serve to retard the angular motion of the latter when the brakes are applied. The assembly of these plates, together with the cords and weights used to clamp them against the drum, is shown on Plates 86 and 88. It will be seen that a small coil spring holds the plates clear of the drum when the brakes are to run free.

The weight carriers and weights are released by solenoid operated triggers, as shown on Plate 89; pistons on the weight carriers work in oil dashpots so as to prevent vibrations of the brake recording pencil, and pistons may be changed so as to give a sudden or gradual application of the brakes as desired.

The cable pull exerted on the model by the brakes is recorded as follows:—The aluminum idler pulley, as may be seen by referring again to Plate 87, carries the bight of the cable which is attached to the model and being unwound from the drum. Any retardation of the drum, as the cable is being paid out, exerts at once a pull on the model, and the combined action of these two forces causes the swinging arm to move. The angular motion of the latter is, however, controlled by the action of two balanced springs at the lower end of the arm; whatever movement takes place is proportional to the resistance of the brakes and is recorded on the paper by a pencil attached to the extreme upper end of the arm. This entire mechanism is accurately calibrated in such a manner that all errors due to angularity, inequalities in the springs, etc., are entirely eliminated.

A short account of the sequence of operations during a launch may serve to explain more clearly the exact method of recording the desired information. Assume, first, that the model has been released and is moving down the ways; the cable is paying out freely as the only resistance is that of the small brush on the chronograph segments.

1. At the designated moment, the controller operates the brake release and allows the weight carrier and weights to drop.

2. The weight carrier, acting through the cords, causes the brakes to grip the revolving drum.

3. The latter, although continuing to revolve and to act as a chronograph, is retarded somewhat by the action of the brake.

4. A pull is exerted on the cable, which, leading around the idler pulley so as to effect a change of direction of 180 degrees, draws the latter toward the model against the action of the double springs.

5. The pencil attached to the arm records the cable pull on the moving paper strip, while a stationary pencil traces at the same time a zero or reference line.

From what has been said in the preceding paragraphs, it will be evident that all elements of the launching conditions may be varied at will, using the apparatus to record all the data for successive series of runs. The various unknown factors are then determined as described below. With regard to the actual performance of the model, it may be said that all parts of the mechanism functioned in a most satisfactory manner and that the results for corresponding and similar runs were remarkably consistent.

To work up the results, it is of course necessary to apply the principles of mechanical similitude as is done for all work in the model basin.

The length ratio, L, being 96, the corresponding speed ratio is $\sqrt{L} = \sqrt{96} = 9.798$. Inasmuch as the maximum launching speed of the model is about 3 feet per second, corresponding to a ship speed of 29 feet per second, and as the vessel in any circumstances would never attain such a high velocity, means are adopted to reduce and regulate this velocity to correspond with what may be reasonably expected on the day of the launch. A short length of cord is attached to the model and drawn through an improvised friction brake, the length of cord being varied to suit the final launching velocity required. As the effect of this retardation is only to reduce the initial velocity, and as this action ceases long before the brakes are applied, it need not be considered except when determining the shape of the entire velocity curve. The final speed (when entering the water) may be varied in this way from 12 feet to 24 feet per second.

The chronograph is so proportioned that each of the intervals between record marks represents about 19.2 feet travel of the vessel. A specimen record is shown on Plate 92. For the sake of convenience, all units, unless otherwise noted, will hereafter be expressed in proper terms for the full-sized vessel. All calculations and curves have been worked up on this basis, as the work is then more easily followed by all concerned.

The length ratio, L, as noted above, is 96.

The displacement ratio is therefore $L^3 = 884{,}736$. For a weight of ship and cradle of 16,000 tons, this corresponds to a model weight of 40.52 pounds.

The ratio of cable pull on the model and brake pull on the ship is also L^3, or 884,736. (From the theory of mechanical similitude, where $f \times s = \frac{1}{2} mv^2$.)

A pull of 500,000 pounds on the ship is equivalent to a pull on the model cable of 9.042 ounces, which, in turn, is represented by an ordinate of 0.81 inch on the recording paper. The model brake is capable of exerting a total relative pull of about 900,000 pounds, or some 400 tons.

The brake mechanism has been calibrated with extreme care, using a specially constructed and calibrated spring to exert a tension upon the cable in its normal direction, with the brakes set and the recording mechanism in operation. After repeated measurements of the resistance, due to the rotation of the drum, the rotation of the idler pulley and the friction of the pencils on the trimming masts, it has been assumed that this resistance may be neglected, as being within the limits of accuracy of the observations as a whole.

In order that the traces of the bow and stern may be correct in shape, it is necessary that the moment of inertia of the model about a transverse horizontal axis through its center of gravity should bear the proper relation to that of the large vessel; that is, K, the radius of gyration, should vary as L. To this end, the model is suspended by two cords of known length equidistant from the center of gravity, and K determined in the usual manner. The inside lead ballast weights are then placed in such position as to fulfil the required conditions. It is understood, of course, that K for the large vessel must be found by more or less approximate methods. The pivoting point, the drop of the bow, the clearance of bow and stern, the final trim afloat and the position of the model when it comes to rest are easily obtained from the record of the trimming masts. The time and point of application of the brakes are indicated on the recording strip as shown on the plan.

For purposes of illustration, there is shown a complete set of curves and data obtained from a set of runs with the model. To facilitate the transfer of all data from the model record to these curves, a set of nomographs has been prepared, giving rapid solutions of about six of the principal equations. The velocity-distance curve, Run No. 119, and the velocity-distance curve, Run No. 121, are drawn directly from the chronograph record. The time-distance curve, Run No. 119, has been plotted as a convenience in checking from the curves of the Cuyama. The curve of brake pull, Run No. 119, is taken directly from the diagram on the record sheet, and the traces of bow and stern pencils from the vertical board shown in the photographs. The contour of bow and stern have been added to make the diagram more complete.

In the figures presented by Mr. Hiley in his paper on the launching brake, the water resistance is assumed to absorb about 20 per cent of the total energy of the vessel. This figure is necessarily quite approximate, for reasons stated at the beginning of this paper, and an attempt has been made at this point to arrive at a more definite value of this quantity. As is well known, the water resistance of the vessel during launching can be represented by the equation $R_w = K_2V^4$, provided the cradle and other fittings are of such shape as to produce wave-making resistance only. This is not exactly the case, however, as the skin friction resistance is a considerable portion of the whole; the water resistance is best represented, therefore, by a simple equation of the form $R_w = K_2V^3$. Although this equation, as it stands, does not conform exactly to the theory of mechanical similitude, it is used here as a means of simplifying the work. To translate model resistances into ship resistances would require an excessive amount of work, if this operation were to be carried out exactly as is done at the Model Basin.

By substitution and integration of the equation $R_w = K_2V^3$, we find that the velocity-distance curve is represented by the following equation, $V = \dfrac{M}{K_2S} = \dfrac{K_1}{S}$ Where V is the velocity and S the distance run, measured from a certain origin, K_1 and K_2 are constants. The value of K_2 for this hyperbolic curve may be found by solving two simultaneous equations representing two points on the velocity curve where the vessel is clear of the ways and running freely.

The curve of velocity thus obtained is only approximate, but it agrees closely with the curve as actually recorded, and it forms a very convenient means of computing the water resistance during the period when the ship is being brought to rest by the brakes. For instance, two points on the curve of Run No. 121 are taken at 19 feet per second and 14 feet per second; the two equations are then $VS = K_1 = 13{,}085$, and $R_w = K_2V^3 = 81.64\ V^3$.

The first equation indicates that the ship, leaving the ways as on Run No. 121, and neglecting the effects of wind and tide, would still have a velocity of 1 foot per second at a distance of 2½ miles.

The second equation enables us to plot the curve of water resistance as shown on the plan, taking values of velocity, of course, from the curve of Run No. 119. When integrated, the curves give the following results:—

Total energy absorbed by water resistance....................	66,740,000 ft.-lbs.
Total energy absorbed by brake resistance..................	138,360,000 ft.-lbs.
Total energy as recorded from curves......................	205,100,000 ft.-lbs.
Kinetic energy of vessel traveling at 19.4 feet per second, when brakes are applied..................................	201,600,000 ft.-lbs.
Difference	3,500,000 ft.-lbs.

Representing an error or discrepancy of less than 2 per cent.

Note that in this case the resistance of the water absorbed about 32½ per cent of the total energy.

Inasmuch as the results shown in the preceding paragraph are obtained by entirely independent methods, the check is very positive and the agreement remarkably close. The curve of brake pull is considered so reliable, that, from data on similar model runs to be made just before the launch, the pressure in the hydraulic brake cylinders and the point of application of the brakes will be determined.

For purposes of comparison, the velocity-distance curve of the Cuyama has been added. The similarity between this curve and the one obtained from the model is most apparent.

It has been necessary, throughout this paper, to comment rather briefly upon what is really a very comprehensive subject, and it has been the intention to supplement this brief description with such photographs and plans as would serve to explain clearly the subject-matter of this article. Unfortunately, at the time this paper was prepared, the launch of the battleship California had not taken place, and it was not possible to make extended comparison of the data obtained from the model and the final data from the actual launch. It is hoped that, when this matter has been worked up, there will be further opportunity to present it to the Society in a subsequent paper.

DISCUSSION.

The President:—The paper by Captain Gleason and Lieutenant Commander Saunders, entitled "The Launching of Large Vessels in Restricted Waters," is now open for discussion. Professor Hovgaard, may we hear from you on this subject?

Professor William Hovgaard, *Member:*—I think any one who has had to do with the launching of ships in restricted waters will recognize the immense value of this paper, where, I believe, for the first time we get connected and scientific data of a launching under these particular conditions.

I have myself felt the need of information of that kind on two occasions. The first was in 1895, when we launched the Imperial Russian yacht, the Standard, built for the Czar.

She was launched at Copenhagen from the shipyard of Burmeister and Wain, under very difficult conditions. It happened that there was heavy ice, 15 inches thick, in the harbor at the time, and we had to cut a channel in the ice a thousand feet long to give the ship free room for launching. We had considerable anxiety whether she would run too far or not. On that occasion I felt the need of information of a reliable and scientific nature from which to judge of how far the ship would run, but we had absolutely nothing, could not find anything, and had to rely on our previous experience with rope stops. As far as this went, it was very good, but this was the largest ship we had ever built in that yard, being more than 5,000 tons displacement, and rather deep, so there was some anxiety connected with it. However, it turned out all right.

The next time I had to do with launching under difficult conditions was in the case of the battleship New Jersey of the Fore River Shipyard. There was used on that occasion the usual rope stops, but in addition there were two brakes, which were made up of a long, wedge-shaped timber dragged longitudinally between two cross timbers, and the friction should absorb a certain amount of energy, but we did not get the full effect in that case. The cross timbers broke, either because the wood was not of good quality or else because the timbers were too light.

In this paper is discussed a proposed new method of drawing wire rope through brakes to give constant friction, and that seems to be a much more perfect method than the wooden friction wedges, because the friction acts through a longer distance, so that a greater amount of energy is absorbed.

There is one point in the paper on page 204 which I think is of great interest, and that is the comparison between the launching of the Kanawha and the Cuyama. The distance run out by the Kanawha was only 329 feet, as against 444 feet in the case of the Cuyama, although the Kanawha was considerably lighter, the difference being due to the stern mask which was fitted. I believe that, of all the means of stopping a ship, there is nothing which compares with the stern mask, because of the enormous hydraulic resistance it offers. It is at once simple and effective. The faster the ship goes, the more effective is the mask, which provides automatically for variations in the frictional resistance under the sliding ways.

I think the authors of the paper, Captain Gleason and Commander Saunders, are to be congratulated on the application of model experiments in this connection and on the ingenuity which they have shown in working out the results and the interesting data they have given us. They arrived, as I did when I had to do with the launching of the New Jersey, at an analysis of the energy consumed. A study of the velocities and accelerations is, of course, of interest, but that is not the kernel of the problem. The main point in an analysis of launching and in designing for new launches is to know what happens to the energy. There is a certain known amount of kinetic energy generated in launching. This energy is reduced by frictional losses, which are also fairly calculable, but when the ship takes the water it is difficult to estimate what becomes of the remaining energy. Evidently the greater part is taken up in ordinary ship resistance, but we know very little about resistance under these peculiar conditions, where the ship goes obliquely into the water, and any analysis of that matter is, therefore, of the greatest interest. To the hydraulic resistance is to be added, of course, the resistance of rope stops or other means of checking the ship. When we get sufficient data of that kind, I think we will be able to launch ships with much greater confidence.

The President:—Is there any further oral discussion? If not, the secretary will read a communicated discussion.

Rear Admiral D. W. Taylor, C. C., U. S. N., *Honorary Vice-President:*—This is a paper of a kind very desirable to have in our Transactions. The launch of a large ship is perhaps the most difficult engineering problem connected with shipbuilding, and when it must be undertaken in restricted waters, the sense of responsibility of the naval constructor and of relief after a successful launching are necessarily much accentuated.

The friction brake methods used in Mare Island launches are certainly sound theoretically, and the fact that they have been used with success on a number of ships shows that the practical application has been carefully worked out. Their extension to the launch of the California is a bold step, but the paper itself shows that it has been taken only after the most thorough and painstaking investigation, justifying the anticipation of the authors of a successful outcome.

Captain Gleason (Communicated):—Professor Hovgaard's discussion is very interesting and is appreciated by the authors of this paper. There is one point which it is desired to invite attention to wherein Professor Hovgaard mentions the comparison between the launching of the Kanawha and Cuyama. The Kanawha had a mask fitted on her stern, while the Cuyama did not. The same type of launching brake was used, and for practical purposes the difference between the distance traveled by the Kanawha and Cuyama may be taken as a fair approximation of the effect of the mask. No definite comparison, however, can be made, for the reason that there were no means in the brakes, as then employed, to register the actual pressure applied to the friction ropes. It is therefore quite possible that the pressure on the friction ropes in the case of the Kanawha may have been greater than in the case of the Cuyama. This would of course account for some of the difference, were it the case. It is regretted, in the case of these two ships, one with a mask and one without, that the hydraulic means of applying pressure to the friction ropes was not used. Had such means been used, there would have been some definite data upon which to arrive at a fairly accurate estimate of the effect of the mask.

Since the paper was read before the Society on November 13 and 14, the California was successfully launched on November 20, using the brakes as described in the paper. The launching brakes functioned satisfactorily, as far as the brakes themselves were concerned, but unfortunately the chains which were used to connect up the friction hawsers to the ship broke at various stages of the vessel's travel, owing to what afterwards was found out to be defective links in the chain. The brakes therefore were not able to produce the effect which had been intended, but sufficient work was done by the brakes before the various chains parted to reduce the speed of the vessel to such a point that two emergency anchors stopped the ship without any damage to her. It is regretted that the breaking of the chain cables prevented the obtaining of valuable data in actual experience for comparison with the model experiment data. From the data, however, that were obtained and the distance that each of the friction hawsers traveled through the brakes before the chain broke, it is hoped that a reasonably accurate estimate of the work done by the brakes may be arrived at. This data will be collected, and it is hoped that at the next meeting of the Society a complete report of the actual performance may be made to the Society.

It may be noted that, in launching the California, chains were used for correcting the friction hawsers to the ship, whereas, in the previous launchings of the Kanawha, Maumee and Cuyama, wire rope hawsers were used. This change was made in the interest of economy in view of the chains being on hand; in the light, however, of the defects that developed in the chains, it would have been preferable to have used wire hawsers, and had this been the case, full working effect of all the brakes would undoubtedly have been obtained. In the preliminary consideration of the launching of the battleship Montana (expected launching weight about 20,000 tons), which follows the California, the idea of using chains has been abandoned and wire rope hawsers will be used, thus eliminating the uncertainty of chains.

THE PRESIDENT:—This is another instance of a paper recording most valuable data, but not lending itself easily to discussion. It is, however, none the less valuable, and shows clearly the painstaking care of its authors. I know that you will desire that the thanks of the Society be extended to the authors for their very admirable presentation of facts.

The next paper on the program is entitled "The Propelling Machinery of the U. S. S. Leviathan," by Mr. E. H. B. Anderson, Member.

Mr. Anderson presented the paper.

THE PROPELLING MACHINERY OF THE U. S. S. LEVIATHAN.

By Ernest H. B. Anderson, Esq., Member.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

This vessel is the second of three large Atlantic liners which the Hamburg-American Steamship Company ordered to be built in Germany during 1911.

The first to be completed was the Imperator, and she made her maiden voyage to New York in 1913.

The original name of the Leviathan was the Vaterland, and at the outbreak of the war in 1914 this vessel had made three round trips between Hamburg and New York and had completed the outboard run of her fourth voyage.

The third vessel was not launched at the outbreak of the war, and it is highly improbable that she will be ready for service for some considerable time.

The Leviathan is the largest vessel ever completed, and without doubt her designers and builders have created a structure that contains many novel and interesting features, most of which have been developed largely from observation and experience gained by studying the designs of the earlier Atlantic liners of this class, namely, the Lusitania, Mauretania and Olympic.

The table on the following page gives the principal dimensions of the vessel, and also comparisons with other vessels of similar type.

It is not my purpose to attempt to describe the interior deck arrangements of the vessel, but during my first visit to this ship I was struck by the fact that the large public rooms and social halls were entirely free of casings, and they had the appearance of being more spacious than those of other vessels. The explanation of this can be seen by referring to Fig. 1, Plate 95, which shows the arrangement of Deck "B" in the Leviathan and Imperator.

In the former, the two sets of uptakes, or the steel casings that carry the gases from the boiler furnaces, are not led to the center of the ship until they have passed through the boat deck, where they join into a breeches piece at the base of the smokestack.

In the latter, the uptakes all join up to the funnel casings at the lower decks and then pass up through the various decks in the middle of the vessel.

The third or the aft funnel does not connect with the boilers in either of the vessels, its purpose being to ventilate the engine-rooms, and in both vessels the casing opens into the engine-room through one trunk at the center line of the vessel.

DIMENSIONS OF LEVIATHAN AND OTHER LARGE VESSELS.

	Leviathan	Imperator	Aquitania	Mauretania
Builders	Blohm & Voss, Hamburg	Vulcan Co., Hamburg	John Brown & Company, Clydebank, Scotland	Swan, Hunter & Wigham Richardson, Wallsend on Tyne, England
Owners	U. S. Government	British Ministry of shipping	Cunard	Cunard
Date of completion	1914	1913	1914	1907
Length overall	950 ft.	905 ft.	901.50 ft.	765 ft.
Length between perpendiculars	....	880 ft.	865 ft.	760 ft.
Breadth	100 ft.	98 ft.	97 ft.	88 ft.
Depth, moulded	63 ft.	62 ft.	64 ft.	60.33 ft.
Gross tonnage	54,190	52,117	46,500	32,000
Draught	38.50 ft.	35.50 ft.	36 ft.	33.50 ft.
Displacement	63,100 tons	57,000 tons	53,000 tons	38,000 tons
Type of engine	Parsons turbines	Curtis Parsons turbines	Parsons turbines	Parsons turbines
Number of shafts	4	4	4	4
Type of boilers	Water-tube	Water-tube	Scotch double-ended	Scotch double-ended
Number of boilers	46	46	21	21 D. E. 2 S. E.
Number of furnaces	138	138	168	192
Steam pressure	235 lbs. per sq. inch	235 lbs. per sq. inch	195 lbs. per sq. inch	195 lbs. per sq. inch
Total heating surface	210,440 sq. ft.	203,009 sq. ft.	138,595 sq. ft.	158,350 sq. ft.
Total grate surface	3,843 sq. ft.	3,763 sq. ft.	3,541 sq. ft.	4,048 sq. ft.
System of forced draught	Howdens	Howdens	Howdens	Howdens
Total S. H. P. of ahead turbines	65,000	61,000	60,000	68,000
Speed in service	22.50 knots	22.50 knots	23.50 knots	25 knots
Revolutions per minute	170	170	165	180

GENERAL ARRANGEMENT OF TURBINE MACHINERY.

The vessel is driven by the Parsons direct drive type of steam turbine arranged on four shafts, and the turbine installation consists of eight separate turbines with their respective rotors.

The go-ahead units, of which there are four, each driving a separate shaft, are arranged to operate in triple series, which is the arrangement adopted in all the large Parsons turbine-driven vessels built after the completion of the Cunard liners Lusitania and Mauretania in 1907.

The go-astern units, of which there are also tour, are arranged to operate in parallel and consist of two independent sets for the two port and two starboard shafts, each unit of which drives a separate shaft. In other words, each shaft with its propeller wheel is driven, respectively, by one go-ahead turbine and one go-astern turbine.

The machinery installation contains many clever features, the chief being that under any extraordinary conditions which may arise at sea it is possible to operate the two turbines of any one shaft entirely independent of the other units. This means that it is hardly possible for any contingency to arise whereby the turbine installation can be entirely disabled.

Further, all the interconnecting valves and pipes between the turbines are located on the lower half of the casings. The design of these valves, together with the hydraulic operation of the valve pistons, is such that any change can be carried out at sea without having to shut down or stop the vessel for any appreciable length of time.

The turbines are arranged in three separate engine-room compartments, as shown on Fig. 2, Plate 96. On referring to this diagram, it will be seen that the four units of the two inboard shafts are in the main or forward engine-room, whilst the two turbines of each outboard shaft are in the two aft engine-rooms, these compartments being separated by the fore-and-aft bulkhead down the center line of the vessel.

The high-pressure ahead turbine drives the port inboard shaft, and coupled directly to the aft end of this is a high-pressure astern turbine.

The mid-pressure ahead turbine drives the starboard inboard shaft, and coupled directly to the forward end of this unit is the other high-pressure astern turbine.

The two low-pressure ahead and two low-pressure astern turbines drive the outboard shafts, each pair being arranged in one of the separate aft engine-rooms.

The forward or main engine-room is about 66 feet long by about 60 feet wide.

The turbine compartment, however, is only 42 feet wide, whilst the feed pumps, feed heaters and other auxiliary machinery are arranged in two side compartments adjoining the turbines, each about 9 feet in width.

The maneuvering valves and gauge board are located at the forward end of this engine-room, the working platform and main floor is directly over the turbines, and nothing is visible of the turbines from this working platform.

The floor does not extend over the two side compartments, and this auxiliary machinery is exposed and can be seen always by the engineers in charge. On this platform or floor, and extending down the center line of the vessel, are located the forced lubrication pumps, oil coolers and oil strainers.

The turbine steam to gland connections, oil supply pipes and valves to the turbine bearings, and turbine drain valves are all operated from manifolds arranged on this platform, and at the aft end the engine-room elevator also opens on to this floor. Directly under this platform gratings are arranged so that access to the rotor bearings and thrusts, and dummy micrometer of high-pressure ahead turbine can be reached quickly.

Floor plates are also provided above the double bottom, over the engine-room bilges, so that the oil drain pipes to the drain tanks are visible. There is also plenty of clear space to allow access to the drain connections under the barrels of the turbine casings.

The wing turbine compartments extend clear across the full width of the vessel and are about 80 feet in length.

No regular platform or floor is arranged in these engine-rooms, but at the same level as the floor of the forward engine-room, gratings extend right round the outboard sides of the turbines.

Access to the turbine rotor bearings and thrusts is provided for by means of ladders.

The air pumps and line shaft bearings of the inboard shafts are also reached by means of a ladder and passageway, arranged beside the entrance door to the aft engine-rooms, and on the starboard side this passageway permits of access to the main dynamo compartments.

Under the regular full-speed-ahead steaming conditions the steam from all the boilers is admitted to the high-pressure ahead turbine, out of which it passes over to the mid-pressure, and from the exhaust end of this unit it is divided and passes into the two low-pressure ahead turbines, and from these to the four main condensers.

For go-astern conditions the boiler steam is admitted to each of the high-pressure astern units, from which it passes over to the low-pressure astern turbines, thence to the main condensers, that is, the port high-pressure astern exhausts to the port low-pressure astern, and the starboard high-pressure astern to the starboard low-pressure astern.

Under maneuvering conditions, such as when entering or leaving port, the two ahead turbines driving the two port shafts and the two ahead turbines of the starboard shafts are operated independently, in which case the mid-pressure turbine takes boiler steam directly.

There are four main condensers of Weir's "Uniflux" design, which have a total cooling surface of 68,700 square feet, two being placed back to back in each aft engine-room, on the inboard side of the low-pressure ahead turbines.

The tubes are ¾-inch diameter external, and the length between the tube sheets is approximately 8 feet 9 inches, whilst the circulating sea water makes two passes.

The exhaust nozzles to each condenser are fitted with two hydraulically operated gate valves about 39 inches wide by 90 inches high, so that any defective condenser can be shut off.

The centrifugal pumps, each pair being driven by one compound reciprocating engine, supply the cold sea water to the two condensers of each engine-room. The discharge pipes from each pump are about 39 inches diameter, and in addition, a cross-connection pipe is also fitted between the pump discharge pipes of each engine-room.

Directly under the condensers there are two single-acting air pumps of Weir's "Dual" type in each engine-room, which discharge to the hot well tanks.

Eight Weir's type of "Simplex" double-acting feed pumps draw water from the hot well tanks and reserve feed water tanks and discharge to the boilers, after passing the water through surface feed heaters using auxiliary exhaust steam.

Six vertical "Duplex" forced lubrication pumps supply oil to the turbine bearings and thrusts.

The two oil coolers are of the double-tube condenser type, the cooling water passing outside the outer tube and internally through the inner tube, whilst the heated oil passes in a film between the two tubes.

In each aft engine-room is also arranged an auxiliary condenser having about 2,100 square feet of cooling surface, together with its circulating water pump and monotype air pump.

A large evaporating and distilling plant is located in a compartment on the port side, outboard of the main turbine engine-room, and correspondingly on the starboard side is the refrigeration machinery.

The main electric lighting plant is arranged in a space aft of the two large wing engine-rooms, and consists of five Brown-Boveri impulse wheel turbo-generators each rated at 445 kw., and 110 volts. This compartment forms a portion of the tunnel for the two inboard lines of shafting and is about 30 feet wide and 66 feet long. An emergency electric light plant, having a Diesel oil engine-driven generator, is fitted well above the water-line and located close to the engineers' quarters.

The tunnel shafts are hollow, about 18⅜ inches external diameter, and the tail shafts 19⅝ inches external diameter.

The two inboard lines of shafting are made up of nine lengths, which are supported by eighteen shaft bearings.

The outboard lines of shafting each have three lengths, with six shaft bearings.

The table on the following page gives the leading dimensions of the various pumps.

DETAILS OF THE TURBINES.

High-Pressure Ahead (*Port Inboard Shaft*).—The body of the casing is made of cast steel, built in four sections, with a permanent vertical joint arranged between the second and third expansion stages, whilst the two halves are bolted together at flanges on the horizontal center line.

Cast-iron pedestals are bolted on to facings at each end of the turbine body and support the rotor bearings, adjusting block, supporting feet to the ship's foundations and the columns of the lifting gear.

The diameter of the rotor is about 8 feet 8 inches and is made up of three separate forged steel drums which have internal flanges and are bolted together.

The two end sections of the drum have other internal flanges at the end of turned cylinders, into which are fitted the rotor end discs, and these are in turn secured by through bolts.

The overall length of the rotor drum, including dummy, is approximately 18 feet (Fig. 3, Plate 97).

Dimensions, Type and Size of Pumps.

Name	Type	No. off.	Dimensions	Strokes per minute
Main air pumps	Weir, dual single steam cylinder	4	21.26″ x 2 x 39.37″ / 24.60″	16
Main circulating pumps	2 comp. recipg. engines combined with 4 pumps		18.50″ x 31.89″ / 19.68″	
	Branches about 39 inches diameter. Impeller 59.06 inches diameter			
Auxiliary circulating pumps	Single cylinder engine and pump	2	8.66″ / 8.66″	
	Branches 10 inches diameter. Impeller 41.34 inches diameter			
Auxiliary air pump	Monotype		10.23″ x 21.26″ / 11.81″	40
Main feed pumps	Weir, single cylinder, double-acting vertical	8	19.68″ x 13.78″ / 31.50″	20
Auxiliary feed pumps	Vertical duplex double-acting	4	15.75″ x 9.84″ / 17.72″	34
Feed heater pumps	Weir, single cylinder, double-acting vertical	8	19.68″ x 13.78″ / 31.50″	20
Oil pumps	Vertical duplex, double-acting	8	11.02″ x 8.68″ / 12″	16.50
Bilge pumps	Vertical duplex, double-acting	4	9.055″ x 10.24″ / 13.78″	32
Ballast pumps	Vertical duplex, double-acting	4	11.81″ x 11.81″ / 13.78″	34
Ash ejector pumps	Vertical duplex, double-acting	4	15.75″ x 9.84″ / 17.72″	34
Bath pumps	Vertical duplex, double-acting	2	14.17″ x 11.81″ / 13.78″	34
Sanitary pumps	Vertical duplex, double-acting	4	14.17″ x 11.81″ / 13.78″	34
Drinking water pumps	Vertical single, double-acting	2	9.84″ x 8.66″ / 9.84″	55
Evaporator pumps	Vertical duplex, double-acting	2	5.98″ x 5.98″ / 5.90″	15
Hold pumps	....	4	6.97″ x 6.97″ / 6.97″	
Brine pumps	Vertical duplex, double-acting	4	6.69″ x 6.69″ / 7.87″	
	In refrigerating machinery compartment at F. E. room			
Circulating pumps	Motor drive centrifugal type	2		
	In refrigerating machinery compartment at F. E. room			

There are 101 rows of standard impulse-reaction blades in casing and rotor respectively, arranged in four expansion stages, with blade heights varying from 3⅛ inches to 6⅜ inches.

The Parsons type of contact dummy is fitted at the steam inlet belt, and the fore-and-aft clearance between the casing and rotor is measured by the usual micrometer. Two steam pipes, each 18⅛ inches diameter, admit steam to the turbine, and the exhaust nozzle is about 42 inches diameter.

The by-pass valves are fitted between the first and second stages, which allow for an overload of about 25 per cent above the designed power.

The rotor shafts are shrunk into the hubs of each disc, and the thrust collars for the rotor adjustment are arranged at the forward end of the rotor.

High-Pressure Astern (Port Inboard).—This casing is bolted rigidly to the aft end of the high-pressure ahead. The body is made of cast iron in two half-sections and bolted together at flanges on the horizontal center line. Cast-iron pedestals are bolted to facings at each end of the turbine body and support the rotor bearings, supporting feet to the ship's foundations and the columns of the lifting gear.

The fixed foot which transmits the propeller thrust through these turbines to the hull of the vessel is arranged between the two casings; thus the high-pressure ahead is free to move forward and the high-pressure astern moves aft, the foot at forward end of the high-pressure ahead and aft foot of the high-pressure astern being arranged to slide freely.

An impulse wheel having three rows of rotor buckets, about 12 feet mean diameter, receives the steam from the regulator valves, and this is followed by four expansion stages of the regular impulse-reaction blading on a drum construction, having a diameter of 9 feet 10 inches. There are 42 rows of impulse-reaction blades in casing and rotor, with blade heights varying from $1\frac{9}{16}$ inches to 5 inches.

A radial fin type of dummy is fitted to the steam belt, and the overall length of the rotor drum is approximately 8 feet 6 inches.

Fig. 4, Plate 97, shows the general design of the rotor drum and impulse wheel. The steam inlet pipe is about 14½ inches diameter, and three sets of nozzles fitted in cast-steel chambers are bolted into openings arranged on the top half casing, each having an inlet branch of 8½ inches diameter, the exhaust nozzle opening being about 29 inches diameter.

Mid-Pressure Ahead and H. P. Astern (Starboard Inboard).—The casing of the mid-pressure turbine is built up in sections made of cast iron, and the two halves are bolted together at flanges on the horizontal center line. Cast-iron pedestals are bolted on to facings at each end of the turbine body and support the rotor bearings, etc.

The high-pressure astern turbine is bolted directly to the forward end of this unit and is similar to the high-pressure astern of the port inboard side.

The fixed foot which transmits the propeller thrust through these turbines is arranged between the two casings, so that the astern casing expands forward and the mid-pressure casing aft.

The rotor has an impulse wheel with four rows of buckets about 11 feet 6 inches mean diameter at the steam inlet end, and this is followed by ten expansion stages of impulse-reaction blades on a drum construction, having a diameter of approximately 10 feet 10 inches. There are sixty rows of impulse-reaction blades in casing and rotor, the blade heights varying from 5½ inches to 13⅛ inches.

A radial fin type of dummy is fitted to the steam belt, and the overall length of the rotor drum is about 17 feet (Fig. 5, Plate 97).

The rotor adjustment block is arranged in the pedestal between the two casings.

When the mid-pressure ahead turbine is operating in triple series—that is, receiving exhaust steam from the high-pressure ahead—the impulse wheel is by-passed and the work done by the steam is entirely through the impulse-reaction drum blading. It is only when this turbine is being operated under "maneuvering" conditions that the impulse wheel buckets are in use, in which case the steam from the regulator valve passes into three sets of nozzles arranged in the top half casing. Two exhaust branches, each 48 inches diameter, pass the steam over to the low-pressure ahead turbines of the outboard shafts.

Low-Pressure Ahead (*Port and Starboard Outboard Shafts*).—These casings are exceptionally large, made of cast iron and built up in a number of sections, and the two halves are bolted together at flanges on the horizontal center line.

The design of these turbines has this unusual feature—the dummy is arranged at the aft end in the main exhaust belt, consequently the interior of the rotor is always full of steam at the pressure of the inlet belt, and this is done possibly to overcome any tendency of the rotor to distortion under varying conditions of temperature.

The dummy is of the regular radial fin type.

Cast-iron pedestals are bolted on to facings at each end of the casing body and support the rotor bearings, etc.

The diameter of the rotor drum is 12 feet 9½ inches, and it is made up of three forged steel drums bolted together at internal flanges. The end discs are also bolted to the drum with bolts.

The overall length of the rotor drum and dummy is approximately 21 feet 3 inches (Fig. 6, Plate 98).

There are sixty-three rows of impulse-reaction blades in casing and rotor, arranged in twelve expansion stages, the blade heights varying from 5¼ up to 24 inches.

The internal diameter of the casing at the exhaust barrel is 16 feet 9½ inches.

The exhaust nozzle opening has an area of 81½ square feet, and directly under this are arranged the fixed feet which transmit the propeller thrust to the hull, the steam inlet branch to each casing being 48 inches diameter.

Low-Pressure Astern (*Port and Starboard Outboard Shafts*).—These casings are bolted rigidly to the aft end of the low-pressure ahead. The body is made of cast iron built up in four sections, a vertical joint being arranged about the middle of the barrel, the pedestals which support the bearings, etc., being bolted to the casing and arranged so that they can slide in the aft direction.

The diameter of the rotor drum is about 10 feet 10 inches, and it is made up in two sections bolted together. The disc ends are bolted to internal flanges arranged at each end of the drum.

The dummy at the steam inlet belt is of the radial fin type and the overall length of the drum is about 11 feet (Fig. 7, Plate 98).

There are thirty-seven rows of impulse-reaction blades in casing and rotor, arranged in five expansion stages, with blade heights varying from $6\frac{3}{16}$ inches to $13\frac{3}{8}$ inches in length.

The steam is admitted to the aft end of the casing through a nozzle about 29 inches diameter.

The area of the exhaust branch to the condensers is about 50 square feet. This branch is built up of steel plates and flanges in five sections of rectangular shape, and forms a large arch from the exhaust end of this astern casing over to the main exhaust trunk (Fig. 8, Plate 98).

Main Bearings.—The lower half of the rotor bearing shells are made of composition, and the design is such that cold sea-water circulates in a jacket through the casting. The surface of the bearing bush in contact with the rotor journal is faced with white metal.

Safety strips are also provided for, so that the blading will not be damaged if the white metal face is melted out.

The dimensions of the rotor bearings are as follows:—

	Diameter	Length on white metal	Length of safety strip
	Inches	*Inches*	*Inches*
H. P. ahead	27½	38½	5
H. P. astern	27½	34	5
M. P. ahead	27½	50½	5
L. P. ahead...........	35½	70¾	6
L. P. astern...	27½	38½	5

The safety strip is arranged at the center of the bearing shell where the lubricating oil is admitted under pressure; grooves for oil are cut in the white metal surface, and in addition four semicircular grooves have been cut on the rotor journals, these being about ¾-inch wide and extending for almost the whole length of the bearing.

Turbine Glands.—The glands have the usual steam supply and leak-off connections and take the regular form of alternate rows of brass fin strips on rotor shaft and gland casing sleeve. At the outer end of each is fitted four Ramsbottom split rings of composition.

Turbine Thrust and Adjustment Blocks.—The turbine thrust and rotor adjustment blocks are arranged as follows:—

In the case of the high-pressure ahead, the collars form part of the forward end of the rotor shaft.

In the case of the mid-pressure turbine and the low-pressure ahead turbines, they are arranged between the two turbines on each of these shaft lines, the thrust shaft being a separate forging having flanges at each end, which are bolted to the shaft ends of the rotors.

Each thrust shaft has ten collars 31¼ inches external diameter and 20 inches diameter at bottom of collars, and about 1⅝ inches thick.

The thrust shoes are secured in cast steel housings and are faced with white metal, the total surface of the bottom shoes being approximately 2,100 square inches in area, whilst the top shoes have an available surface of about 1,900 square inches.

The adjustment of the rotors is made in accordance with Parsons' usual practice. To assist in the adjustment of the rotors, a special device is fitted which enables the engineers to freely move the heavy rotors and the lines of shafting.

A cast-steel bracket is secured to foundations under one of the shaft couplings in each of the tunnels. At both ends of the bracket a square extension projects from screw-cut threaded spindles, the nuts of which carry ball-bearing mounted steel rollers which can be screwed hard against the flange couplings. By using this, and at the same time having the turning gear in operation, the rotors can be moved easily (Fig. 9, Plate 99).

Impulse-Reaction Blading.—The standard Parsons' blade sections are used throughout, the material being a common brass mixture; the groove widths varying from ½ inch to 1½ inches, the width of the blade section being dependent on the length of the blade.

The usual method of fitting separate packing sections between each blade is adopted.

About the center of the forward edge of each groove in casing and rotor, a semicircular projection is left, and each blade and packing section is correspondingly notched out so as to fit over the projection (Fig. 10, Plate 99).

By this method each blade and packing section is held mechanically. The binding strips are of round brass wire, which is fitted in a hole drilled through the blades, and the wire is then silver soldered to each blade. In the long blades there are three rows of binding wires. The ends of all the blades are thin tipped, in accordance with Parsons' usual practice.

Impulse-Wheel Buckets.—The composition buckets of the impulse wheels are fitted similarly to those of the reaction blading, that is, the buckets and the packing sections are separate. The method of holding them in the rotor grooves is, however, slightly different.

The rectangular grooves of the rotor have semicircular projections on each side, and the buckets and sections are notched out to fit over these projections (Fig. 11, Plate 99).

To allow for fitting, at one place on the rotor rim the notches are cut out and the buckets and sections entered at this point and driven round. The tips of all the rotor buckets are thinned and no shroud strip is fitted, which is somewhat unusual.

In the case of the high-pressure astern impulse-wheel buckets, the first and second rows are entirely unsupported, but the third row has a round wire threaded through each bucket and silver soldered; the widths of the buckets are about 1 inch, the lengths varying from 2⅛ inches to 3½ inches.

The mid-pressure rotor has four rows of buckets on the impulse wheel, the first row is entirely unsupported, the second and third rows have one round wire threaded through each bucket and the last row has two round binding wires. The lengths of the buckets vary from 4 inches to 7½ inches and are about $1\frac{3}{16}$ inches in width.

Water-Gauge Glasses on Turbine Casings.—On the lower half of each turbine casing, at the steam inlet belt and at the exhaust belt, a water-gauge glass column is fitted so that the engineers can see when there is water lying in the bottom of the casings.

Turbine-Lifting Gear.—The gear for lifting the upper half casings and the rotors is operated by electric motor, one motor being arranged to do the work of lifting the parts of the two turbines of each line of shafting. The motors, counter shafts, worms and worm wheels are all secured to brackets in the roof girders of the engine-rooms.

Each casing has four large, square, thread-screwed columns, one of which is fitted at each corner of the turbine; the lower end of each column is let into a recess in the cast-iron pedestal and has ball-bearing surfaces.

The threaded columns are each fitted with a composition nut which is arranged to fit into lifting pads cast on the top half of the casing. The upper end of the columns have flanges which are bolted to the gear from the rotor. The threaded columns revolve, and in this way the top casing is lifted evenly and always remains parallel to the joint.

The rotors are lifted in the same way except that a cast-steel girder is fitted between the two columns at each end of the turbine, and to carry the rotor a strap is fitted under the rotor shaft and bolted to the girder. The girder is made to fit over the composition nuts, and the rotor is then raised right up into the top half-casing.

Shaft-Turning Gear.—Each line of shafting is fitted with electric motor turning gear, which is arranged on the aft bulkheads in the turbine engine-rooms.

The Maneuvering Gear.—The entire maneuvering of the turbine engines is carried out from the working platform of the forward engine-room, the regulator valves being all arranged at the forward bulkhead.

To give protection from the heat radiated off these large high-pressure steam fittings, a vertical steel plate screen has been erected in front of the valves, about

7 feet clear from the bulkhead, and on this plate are mounted the valve wheels, pressure gauges, etc.

Directly over these hand wheels, on the roof of the engine-room is a diagrammatic model of the turbines, each unit of which is lighted up by electric light lamps, under the different methods of operation (Fig. 12, Plate 100).

For instance, in "Triple-Series Connection" the words "Triple-Series Connection" show a white light, and all the rest of the model is dark. Immediately the high-pressure regulator is opened the four ahead turbines are lighted up, and, if this order came from the chart house or bridge, then the word "Ahead" is also lighted up. When maneuvering, the words "Maneuvering Connection" show a white light, and, if the port ahead engines are to be used, the bridge signal lights up the port "Ahead," and when the starboard astern signal is given the word "Astern" is lighted, and on opening this valve these engines show a red light. Again, for independent operation, the words "Single Connection" are lighted up, and on opening any of the regulator valves that engine is immediately lighted.

Four lines of steam pipes pass the steam from the boilers over to the engine-room, two on the port side and two on the starboard side. The stop-valves connecting with these steam lines are in the engine-room, and they are operated hydraulically.

A cross-connection pipe runs between the port and starboard valves. From each of these valves the steam passes into a steam strainer chest combined with an automatic trip valve, which has connections to the five main regulator valves of the turbines.

Fig. 13, Plate 101, is a photograph showing the arrangement of the hand wheels. Commencing on the port side they are as follows:—

1. Port high-pressure astern.
2. Port high-pressure ahead.

These two valves connect directly to the port side boiler stop-valves.

3. High-pressure ahead, from starboard side stop-valves.
4. Mid-pressure ahead.
5. High-pressure astern.

Fig. 14, Plate 102, illustrates the general design of the turbine regulator valves.

The by-pass valve is operated by the smaller hand wheel, arranged on an internal spindle passing through the stem of the main valve. The hand wheels are connected to the valve spindles through miter wheels and ordinary spur gears.

The six hand wheels on the lower row admit steam to the low-pressure ahead and low-pressure astern turbines when these are being operated in single connection, the two end wheels being shut-off valves to these lines. These valves are only used in an emergency.

The five small wheels arranged below the others operate the hydraulic cylinders of the change valves. The central one connects to the double change valve of the mid-pressure turbine (Fig. 15, Plate 102).

Fig. 13, Plate 101, shows the various water pipes connecting with these hydraulic cylinders. Above each will be seen a small rectangular box having an electric light and name plate stating clearly what position the change valve is in.

No interlocking gear is fitted between the ahead and astern valves, and this appears to be unnecessary in view of the precautions taken to ensure the proper operation of the turbines.

On again referring to Fig. 13, above the second and third wheels (steam to high-pressure ahead), a small hydraulic cylinder will be noticed, the purpose of which is as follows:—

Under "Maneuvering Connection" it is essential that not more than one-half of the steam from the boilers should be allowed to pass through the high-pressure ahead and port low-pressure ahead turbines, otherwise these engines could be heavily overloaded. On switching the change valve into maneuvering connection, a cam connected with levers opens valves operating the two hydraulic cylinders and locks the high-pressure ahead valve from the starboard side steam lines, and the other valve will only open a limited amount.

Interconnecting Pipes and Change Valves between the Turbines.—The different ways in which the turbines can be operated and the means taken to do this have features of interest. Dealing with the high-pressure ahead, this unit can be operated in three different ways:—

1. The exhaust steam can pass over to the mid-pressure turbine, as in "triple-series" connection.

2. It can exhaust directly to the port low-pressure ahead, as under "Maneuvering."

3. The exhaust steam can pass directly to the condensers of the port side.

Fig. 15, Plate 102, illustrates the change valve, the forward chest of which is bolted to the inlet branch of the mid-pressure and the aft chest to the inboard nozzle at the exhaust steam belt of this casing. The valves are of the piston type, hydraulically operated, and the two views indicate the way the steam flows under the different methods of operation. Another change valve of similar design (port No. 2 on engine-room arrangement, Fig. 2, Plate 96) either allows the steam to pass over to port low-pressure ahead or to condenser. This valve is operated hydraulically from the working platform. A gate valve is fitted on the pipe opening to the condensers; this is worked by hand and is always kept closed.

The mid-pressure ahead turbine can be worked as follows:—

1. Triple-series, receiving exhaust steam from the high-pressure ahead.

2. Maneuvering; live steam is admitted to the nozzles of the impulse wheel, passes through the reaction blades and over to starboard low-pressure ahead.

3. The exhaust steam can pass directly to the condensers of both port and starboard sets through the change valves on port and starboard side (No. 2).

The high-pressure astern turbines of the inboard shafts can be used as follows:—

The exhaust steam from each of these units can pass over to the low-pressure astern, or, for emergency conditions, change valves are installed (No. 3, Fig. 2) so that they can exhaust direct to the condensers. The exhaust-pipe connections between the astern units are exceptionally long and about 29 inches in diameter. They are arranged on the lower half of the casings, and on the starboard side the length of this pipe is about 130 feet and on the port side about 100 feet.

MAIN TURBINE REPAIRS AND OVERHAUL.

At the time the vessel was taken over, the high-pressure ahead turbine casing was open. The rotor had also been lifted, and on examining the blading, dummy, glands and bearings, it was soon seen that this engine was in perfect condition, and in a short time the rotor was lowered and the casing closed up for good.

The manhole on the aft end of the mid-pressure was opened up and a preliminary examination appeared to indicate that this engine was also in good working order.

The rows of impulse wheel buckets were exposed by opening a manhole on the upper half casing. Manholes arranged at each end of the low-pressure ahead turbines allowed me to thoroughly examine the end rows of blading, and this was all found to be in good order.

The safest course to follow as regards finding out the internal condition of these three ahead turbines was to turn them with the turning gear in both ahead and astern direction and listen carefully for unseemly noises. Steps were taken to do this, but it was found that the cast-iron brackets which support the various parts of the turning gear were badly fractured. Repairs were put in hand at once, and on July 10 the mid-pressure rotor was revolved in the ahead and astern directions and found to be in splendid working order.

The preliminary examination of the four astern turbines showed that a considerable amount of damage had occurred in the blading.

Both high-pressure asterns showed damaged blading at the exhaust end rows, and in addition a pocket about 18 inches wide had been cut through ten rows of rotor blades and ten rows of the casing blades on the port turbine, and six pairs of rows on the starboard. The condition of the blading at the steam inlet end could not be seen until the upper half casing was lifted.

Both low-pressure asterns also showed that there was damaged blading at the steam inlet end and also at the exhaust end, and in addition pockets about 15 inches wide had been cut through nine pairs of rows of blading in one case and seven in the other.

High-Pressure Astern (Port Inboard Shaft).—Upon lifting the top casing and rotor, the following conditions were noted:—

(*a*) The three rows of impulse wheel buckets of rotor were badly damaged and required to be entirely renewed.

(*b*) The cast-iron dummy cylinder had been removed, and this was found lying around badly fractured.

(*c*) The dummy piston of the rotor was seriously distorted, and pieces of the dummy cylinder and the brass fin strips had fused in lumps on to the surface of the piston.

(*d*) The four last rows of the rotor blades and three rows of top and bottom casings, together with the blading cut out in the pocket, all required renewal.

The following repairs were made:—

A new cast-iron dummy cylinder was fitted. This was made in six pieces, three for the lower half and three for the upper.

Attempts were made to chip, file and grind the dummy piston of the rotor, but this was not satisfactory, and eventually this was machined and new dummy fin strips fitted. To machine the dummy piston, the rotor was lowered into place and connected up with the turning gear, a cut was taken across the surface and the grooves for the fin strip were turned out.

All the damaged blading at the exhaust end of rotor and casing was renewed, the total number of new blades being approximately 6,500.

After careful consideration it was deemed advisable not to refit the impulse wheel buckets, inasmuch as the indications tended to show that the blading troubles were caused by distortion of the impulse wheel under steaming conditions. Therefore the three rows of buckets in the rotor wheel were removed, together with the two segments of casing guide vanes, as well as the nozzles in the cast-steel chambers.

This turbine was opened up early in June and closed up on October 11, 1917.

Low-Pressure Astern (*Port Outboard*).—When this turbine was opened up, the following blading repairs were found necessary:—

Rotor.—(*a*) The first four rows and part of the fifth row at the steam inlet end were damaged.

(*b*) First row of second expansion stage.

(*c*) The last two rows at the exhaust end, together with a pocket 15 inches wide cut through nine rows of blades.

Lower half casing.—The first four rows of the steam inlet end and the first row of second expansion all badly damaged.

Top casing.—The first four rows at the steam inlet end, the first row at the second expansion, and part of the last two rows at the exhaust end were damaged. In addition to this, a pocket 15 inches wide was cut through eight rows of blades.

All this blading was refitted completely, amounting in all to about 13,500 blades and packing sections, and the turbine was closed up on September 25.

On October 7 this turbine was coupled up to the low-pressure ahead and both revolved by the turning gear in ahead and astern direction, and everything was found to be in order.

Low-Pressure Astern (*Starboard Outboard*):—

Rotor.—(*a*) The first four rows at the steam inlet end and the last row at the exhaust end were badly damaged.

(*b*) A pocket about 16 inches wide had been cut through the last seven rows of blades at the exhaust end.

Lower half casing.—The first four rows at the steam inlet end were badly damaged.

Top casing.—(*a*) The first three rows at the steam inlet end and last row at exhaust end were badly damaged.

(*b*) A pocket about 16 inches wide had been cut through seven rows of blades.

All this blading was refitted, amounting to about 10,000 blades and packing sections.

This turbine was closed up on October 4, and on October 20 it had been coupled up to the low-pressure ahead and both revolved in ahead and astern directions satisfactorily.

High-Pressure Astern (*Starboard Inboard*).—When this casing was opened it was seen that very considerable repairs would be necessary to make this turbine work successfully:—

(*a*) The three rows of the impulse wheel buckets of rotor were badly damaged.

(*b*) The dummy cylinder had been removed, and the dummy piston showed clearly that it had been rubbing in contact with the dummy cylinder. Lumps of metal had fused on to the piston surface and it was badly distorted.

(*c*) The four last rows of rotor and casing blades were damaged and required renewal, together with a pocket 18 inches wide which had been cut through six rows of rotor blades and six rows of casing blades in the lower half.

(*d*) The cast-iron casing was also found to have large cracks or fractures, which had undoubtedly been caused by the revolving dummy piston of rotor coming in contact with the dummy cylinder, which eventually fractured into a number of pieces. These pieces apparently jammed whilst the rotor was revolving, and finally resulted in fractures developing in the main casing walls.

Top casing.—(*a*) On the interior surfaces there were four fractures in all, three being in the nozzle openings. Fig. 16, Plate 103, shows the approximate position of the damage in the casing, and also the extent of the fractures.

(*b*) Externally, there were small cracks under the nozzle openings, in line with the internal fractures.

Fig. 16A is a section across the inlet steam belt of the top casing and looking forward.

Fig. 16B is a hand sketch of a cross-section, in way of the nozzle openings.

Lower casing.—In this half of the casing the damage was much greater, and the question of making satisfactory repairs became a very serious matter.

(*a*) There were eleven internal fractures or cracks, and Figs. 17A and 17B, Plate 103, give the approximate location of the damage and also show the extent of each fracture.

(*b*) Fractures 1 to 4 are somewhat similar to the three in the upper half; they extend from the facing to which the dummy cylinder is bolted right over to the dished end wall of the casing.

(*c*) Fractures 5 to 8 occurred in the wall which supports the end of the dummy cylinder, and are similar to fracture 4 of the upper half casing.

(*d*) Fractures 9 to 11 are across the facing to which the dummy cylinder is bolted.

Around the external wall of the lower half casing eight bulb webs are arranged at equal distances and support the dished end of the casting. Seven of these were fractured completely through, and the damage extended into the main body of the cylinder wall.

The engineers had attempted to limit the extent of the fractures, for brass pegs had been fitted at the end of some of the cracks.

A drain facing on bottom of the turbine was also cracked at the base for about one-half of the circumference, and pegs had been fitted to each end of this fracture.

Fig. 18A, Plate 104, shows an external end view of the lower casing and approximately indicates the extent of the damage.

Fig. 18B, Plate 104, is a side elevation of the casing showing the approximate shape of one of these bulb webs.

A small coffer dam was built up in the casing at the steam belt and filled up with water, but the water began to pour through to the outside, showing that the fractures extended through the full thickness of the casing wall, which is about 3½ inches thick.

Upon reporting the conditions to Capt. Earl P. Jessop, U. S. N., Engineer Officer at the New York Navy Yard, who was in charge of all the repairs in these ships, Mr. Jessop made a thorough examination of the casing and strongly advised electric-welding the fractures. Mechanical fitted patches could not be fitted internally, because the damage was all located in the casing where it supports the dummy cylinder.

Mr. Jessop had practically completed the repairs to the reciprocating engines of two ships by this time, and all the damage had been made good by electric welding. A careful inspection of this work was made and showed conclusively that electric welding would make a thoroughly efficient repair.

Mr. Wilson, of The Wilson Welder and Metals Company, also reported most favorably as regards bringing this work to a successful completion, and about the middle of July work was commenced on preparing the fractures for welding. This was carried out as follows:—

Along the line of fracture a V groove was chipped out. On either side of this groove ¾-inch diameter holes were drilled and tapped, into which were fitted steel studs, the end of each projecting above the metal about ¼ inch to ⅜ inch. In most cases two rows of studs were fitted and staggered, the pitch being about 2¼ inches.

The welding material used was a steel alloy wire, about $\frac{3}{16}$-inch diameter. The first layer was placed along the length of the fracture at the bottom of the V groove. This layer was then peaned down; if found to be loose, it was cut out and the work begun over again. On top of this, another layer was welded in and again peaned down, and this process was repeated until the weld was built clear around the studs and projected about ¼ inch above the surface of the adjoining cast iron.

These repairs progressed slowly, but the conditions under which the men were working were exceptionally trying. It was in midsummer, and only two men could work in the cylinder casing at one time. They had to work doubled up, with their knees touching their chins, because above them was the rotor with its rows of jagged blades. It was not possible to move this out of the way, and the men were also breathing continuously the fumes from the electric arc.

To repair the top half casing, the rotor was lowered and the work was again done in very hot and close quarters.

Whilst the electric welding was in progress, practically no other repair work could be carried out, owing to the glare from the arc blinding anyone standing by.

After this work was completed the casing was tested under water pressure to about 100 pounds per square inch, and found to be practically watertight.

Before the new dummy cylinder could be fitted it was necessary to grind down the weld patches until they projected only about ⅛ inch above the faces of the adjoining wall, and the dummy cylinder castings were notched out to fit over the patches.

All the damaged reaction blades were replaced; this amounted to about 8,000 new blades and packing sections, and the turbine was closed up on October 31.

Engine-Room Ventilation.—Two large fans are arranged at the after end of the aft engine-rooms and supply air through ducts to the main engine-rooms.

Two of the Howden forced draught fans for the boiler-rooms are installed at the forward end of the main engine-room and ventilate the hottest part of the machinery spaces.

To ventilate the closed-in spaces under the platform of the main engine-room, the floor forms an air duct about 10 inches deep, and cooling air is circulated to various parts around the turbine bearings, etc. This method also helps to keep the working platform cool.

Boilers.—The boilers are arranged in four watertight compartments, three compartments having four groups with three boilers per row athwartships, whilst the forward compartment contains ten boilers, making a total of forty-six boilers in all.

Coal bunkers are arranged in the wings at the sides of the boilers, separated by watertight compartments.

The boilers are of the Yarrow type, having one steam drum connecting through two groups of tubes to the lower drums, each boiler having three furnaces. The working pressure is 235 pounds per square inch gauge, and the heating surface of the generating tubes is approximately 210,000 square feet. They are designed to work with Howden's forced draught, four large breast fans driven by compound reciprocating engines supplying the air.

The boilers were found to be in fair condition, and only two required re-tubing.

The brickwork was renewed in a large number, and it was also found necessary to install new fire bars, so as to allow more air to pass through the fires.

The methods used for draining the steam lines from the boilers were very poor, the only provision being a small drain cock and pipe led from the stop-valve on each boiler down the boiler front to the bilges.

The boiler installation is very similar to that of the Imperator, and a full description, with illustrations, was published in *Engineering,* June 12, 1914.

General Overhaul.—Whilst the repairs to the main turbines were in progress, the auxiliary machinery was overhauled thoroughly; the four main condensers were water-tested, new tubes fitted to the auxiliary condensers, and new packing was fitted to the gland bushes of all reciprocating engines and steam cylinders of the auxiliaries.

The main steam lines, as well as the auxiliary steam lines, were fitted with proper drains and traps.

The forced lubrication system, including oil coolers, oil drain tanks and piping, was thoroughly cleaned.

The line shaft bearings were removed and freed of all grit and dirt, and the stern tube glands repacked.

DOCK TRIALS.

The preliminary steam trials of the turbines began early in November and were made with the turbines uncoupled from the line shafting.

All the turbines were driven up to full-speed revolutions, namely, 180 per minute, in ahead and astern directions.

A slight hitch occurred with the starboard high-pressure astern. The regulator valve jammed open, resulting in the turbine gaining momentum rapidly. On shutting the by-pass valve the steam pressure on top of the piston forced this down with a bang, and on opening out the valve it was found that the piston valve was fractured right around the metal which supports the valve stem. It is useless to speculate just when this fracture took place, but it is not improbable that it may explain why this turbine was damaged much more than the other units.

SEA TRIALS.

The vessel left New York Harbor for extended trials on Saturday, November 17, and returned on Thanksgiving Day, November 29. During that time various trials were carried out at cruising speeds, and records of the coal consumed were made.

A short trial was made at full speed, the average revolutions of the four shafts being 171 per minute, corresponding to a speed of about 22.50 knots. No torsion meters were in use, but the estimated shaft horse-power amounted to about 65,000.

The main turbines worked splendidly throughout the whole series of the trials, and absolutely no adjustments had to be made. The engines maneuvered splendidly, and the astern turbines did good work.

Minor adjustments were made to some of the auxiliary machinery and to the steering gear, but, taking everything into account, the performance of the machinery was most satisfactory.

In conclusion, I should like to take the liberty of adding a few remarks in regard to the personnel who carried out the overhaul and made the necessary repairs to the machinery, and to the naval men who operated this machinery during the twenty round trips across the Atlantic.

I consider the successful completion of the overhaul and repair work is largely due to the splendid organization that Capt. Earl P. Jessop, U. S. N., got together in the early stages.

Mr. William H. Mackay, marine engineer, was appointed chief engineer when the vessel was first taken over, and under his direction, within a short time, steam was raised on two boilers, a turbo-generator was put in commission, and very soon the engine-rooms began to look and feel normal, and not have the appearance of cold, damp tombs. Further, the vessel was lighted throughout by her own power.

Mr. Mackay personally superintended the opening out of all the machinery, and we had to act very cautiously in making the preliminary internal examinations, for no one ever felt quite sure that we would not run into some form of booby trap.

The brunt of the actual repair work was borne by mechanics from the New York Navy Yard, Brooklyn, and words will never explain just what great work these men carried out. Strikes occurred from time to time, but during the entire overhaul the navy yard men stuck to the job, and the foremen of the engineering department had the satisfaction of seeing their labors brought to success on the completion of the sea trials.

As regards the naval engineers who operated the machinery, the work of the chief engineer, Commander V. V. Woodward, U. S. N., is far beyond praise. I can only hope that he will receive from the Government the award he has earned so splendidly.

Everyone knows what great service the U. S. S. Leviathan has done during the great war, and the personal efforts of the chief engineer of this vessel are largely responsible for this fine achievement.

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DISCUSSION.

THE PRESIDENT:—I am sure that we have been much interested in this presentation of Mr. Anderson's paper, "The Propelling Machinery of the U. S. S. Leviathan." It is hardly necessary to say that the repairing of the Leviathan's engines was really one of the most notable mechanical achievements of the war. The presentation by Mr. Anderson has been most interesting, and I trust that there will be those here who will desire to supplement the information he has given. I was hoping that there might be present someone who had actually been on the ship at sea since the repairs were made, but apparently there is no one with that experience present

MR. W. H. MACKAY, *Member:*—There is a lot of detail in reference to the Leviathan that it would be impossible to put in the paper as presented by Mr. Anderson. I happen to be one of the men and the chief engineer in charge of the work under Captain Jessup, U. S. N., and I wish to relate to the members of the Society some of the difficulties we ran into.

In the first place, the vessel was taken over on the 7th of April, 1917. I was assigned to her by Captain Jessup on the 12th, after she had been turned over to the Shipping Board by the Collector of the Port of New York. I was under the authority of the U. S. Shipping Board. When we went aboard that vessel all of the lights were out, everything was dark, and the ship was as cold as an iceberg. It was necessary to go through the ship very carefully with flashlights. After we had made an inspection and ventilated the ship, which took us seven days, we started to trace out the piping. There was not a scrap of paper of any kind on board that would give us any information as to her equipment.

I mention these items because it is only fair that the engineers of the American Merchant Marine who were engaged on the work on that ship should be remembered. Every man in charge of the work under me was a licensed engineer under a license issued by the British or the American government, and every one of those men was heart and soul in his work, due to the fact that it was said by the Germans in Hoboken that that ship would never leave the dock, that she was 27 feet in the mud and had 10,000 tons of water in her, and they would never move her, but nevertheless these engineers completed the work, and in three months and twelve days we had steam on her and did not have a scrap of paper to guide us.

We searched the ship night and day for fourteen days before we started repairs, and in opening and closing doors we did not know when we might bid good-bye to this world and visit some other place. There was one instance, which I recall, when our friend Mr. Anderson disappeared for four or five hours in the afternoon. I became worried about him, and after calling him for an hour and searching for him with a flashlight, I heard a voice coming from under the engine-room platform answering, "Billy, I am here." He was down under the turbine searching with a flashlight, tracing out the piping, and it was half past six that evening before he came up. (Applause.)

In connection with that work there are three other things which should be remembered. This is the first time that anything has been permitted to come out about the ship. Captain Jessup said to me one afternoon in my cabin:—"What are we going to do about blading for the ship and men to reblade?" I suggested that we get in touch with the Westinghouse Electric & Manufacturing Company and see if they could not help us out. I called up the officials of the company in New York and explained the situation, and then immediately got in touch with their Pittsburgh Works. Pittsburgh said:—"We will send you twelve men on a special Pullman; we will take the men off our regular work right away." The men left that night and were in New York the day after, but one, coming in with their tools and starting to work. The General Electric Company was notified of the conditions. Mr. Rice called a special meeting of the Board of Directors in New York, and they sent word to Schenectady. Another Pullman car came through with twelve more men. That is the way the work started on reblading these engines.

The next thing was to get the blading. The Coe Brass Company of Connecticut took a sample of the material of which the blades were made, made an analysis, and then they notified us on the telephone:—"We will shut down the works for two weeks and do nothing but make these blades." During that time the tools for boring, blading, and putting the blading into the ship were made in New York Navy Yard under the supervision of Captain Jessup, U. S. N. All the material came to the ship in good shape, and the Coe Brass Company is entitled to the best thanks of the citizens in the country for the work they did in this matter.

I merely mention these facts so that you will understand that every man brought in contact with the undertaking was heart and soul in the work, and every one connected with it did everything he could to make the work a success. The fathers and mothers of the boys taken across are happy that their men were carried across on that ship, and that the vessel arrived safely.

Mr. Anderson:—It was not an intentional omission on my part that I did not mention everyone connected with the work. I found it impossible to refer to all the firms and people who did help to bring this work to a successful issue. I can assure the members that everyone who became connected with carrying out the repairs did everything possible, and the final result proved this conclusively.

Mr. John Platt, *Member:*—I want to make one remark with regard to the fine work that was done. You saw from the diagrams, and what Mr. Anderson pointed out to you, the extremely complicated system of piping nd valves, and everything else that went with that layout. When the ship was turned over to the United States authorities, the German chief engineer on board the ship made the statement to those in charge:—"You will never run her"—and he meant it. He had gone over the whole situation. He had all the drawings with him and the fullest kind of information from the builders, but he had not been able to successfully run the ship himself. On the three round trips she had made they had smashed up the inside of the reversing turbines and done a lot of damage of that kind, and they could not operate the ship. He was of the opinion that the men here would never be able to do it, and so stated to them.

I thought this might be of interest in connection with the complicated work that was done.

Mr. W. L. R. Emmet, *Member of Council:*—I think it would be interesting if Mr. Anderson would give us an idea how the damage was produced.

Mr. Anderson:—The primary cause of the damage to the high-pressure astern turbines was due to frictional heat generated by distortion, causing the revolving rotors to come in contact with the fixed cylinder casing barrel.

There have been various statements made to the effect that the damage was wilful and deliberate, but after making a thorough examination of the turbines I became convinced that this was not so.

The impulse wheel is subjected to extreme temperature changes and a certain amount of distortion is inevitable, especially when steam is turned on quickly and in large volumes. The design of the impulse wheel is such that, if there is any distortion, the dummy piston is affected, because this piston is directly attached to the wheel disc by studs, and even a slight distortion is to some extent magnified at the forward end of the dummy piston. (See Fig. 4, Plate 97.) In other words, distortion of the impulse wheel affected the revolving dummy piston, which caused this to come in contact with the dummy cylinder, with the result that great heat was generated, finally causing fractures in the cast iron, and in turn this produced damage to the wheel buckets.

The axial clearance between the buckets of the impulse wheel and the guide buckets attached to the cylinder casing appears to be somewhat on the small side, and this allowed the moving and fixed buckets to come in contact.

The cast-iron dummy cylinder broke in both port and starboard turbines, and the broken pieces apparently jammed against the revolving rotor, causing much damage and creating intense vibrations and generating tremendous heat.

The damage to the blading at the exhaust end of both turbines was probably caused by vibrations, set up when the broken pieces of the dummy cylinder jammed against the rotor.

We overcame all troubles with these turbines by removing the buckets of the impulse wheel and by taking out the nozzles, which allowed the high-pressure steam to immediately fill up the large inlet belt—or what is the impulse wheel chamber—and this evenly heated the casing.

Dealing with the low-pressure astern turbines, it is quite probable that this damage was caused by blading material being carried over into these turbines from the high pressure astern.

The German engineers fully realized that there was something seriously wrong with these astern turbines, but apparently they did not know just what to do, or they may not have had time to do anything whilst the ship was in port preparing for the following trip. They had cut passageways through the blade rows about 16 inches wide, and it is difficult to understand just what they expected to gain by doing this.

Mr. Emmet:—I had been told about the channels which were cut through the blading, and it seems to me this was a matter of intentional damage.

Mr. Anderson:—I believe this was done with a view to finding out the extent of the damage in the blade rows, and I cannot see what would be gained by any deliberate or intentional damage of this nature. It would have been better practice to cut away the entire end row, and after this was done the next row would be fully exposed, and if this showed damage then it in turn required cutting out, and the engineers would have found good blading and a rotor that would turn freely after cutting away four pairs of rows. In one turbine we did find that a chisel had been driven right in among the blade rows, and it would appear that one of the men had done this more in spite than for any real purpose. I think he was probably starting to cut a channel way, but found it impossible to wield a hammer, so he drove in the chisel and left it there. It was about 16 inches long.

It should be noted that the rotor buckets of the impulse wheel had unusual features. The first two rows were entirely unsupported, *i. e.,* they had no shroud and no support other than that of the root fixing. The third row had a wire strung through each blade about half an inch down from the tip, and the blades were soldered to the wire. In addition, the blade tips were thinned similar to Parsons practice for reaction blades. The guide blades in the casing were also unsupported. (See Fig. 11, Plate 99.)

Mr. Emmet:—Were any of the impulse wheel buckets broken off, or had any fallen out of the rotor wheel?

Mr. Anderson:—Absolutely none were broken out. There was no sign of a broken-off bucket, and the root fixing was a splendid job. The discharge edge of the nozzles were pretty well choked up and mashed, and many of the buckets in the wheel were twisted and bent badly.

When we first of all tested the starboard high-pressure astern turbine under steam, we found that the throttle valve would not close after being opened, and luckily we escaped having a serious accident with this turbine, because the chief engineer, Mr. Woodward, U. S. N., immediately shut off the bye-pass valve, and this forced the piston down and shut the steam off. On opening up the valve it was found that the body of the cast-iron valve had been broken right round the part where it joins the valve stem, and whilst this permitted the valve to open, it would not close. (See Fig. 14, Plate 102.) It is quite possible that this valve was broken whilst the ship was in service, because on the first trip out of New York Harbor the ship was driven right across the North River and almost crashed into the piers in Manhattan.

With reference to Mr. Platt's remarks, I have copied a short article which was published in the *New York Herald* of February 3, 1919, and again on referring to my remarks regarding the accident to the ship when backing out of the dock in Hoboken, on starting the return trip of her first voyage, I have copied an article from the *New York Sun* of May 27, 1914, which articles are as follows:

(From the "New York Herald," February 3, 1919.)

U. S. WORKED MIRACLE WITH VATERLAND

The government officer who convoyed the chief engineer of the old Vaterland to his first internment on Ellis Island asked him what he had done to try and ruin her engines, according to an article in the *Red Cross Magazine.* He in return was equally frank, even though he laughed in his captor's face.

"Ruin her?" he roared. "Mein Gott, she was ruined before she finished her first trip across the Atlantic. There never has been a time since she first touched the water when she hasn't been ruined. I will take off my hat to any American engineer who will ever get her across the ocean again."

He spoke the truth. The Vaterland, as her German builders had turned her out, was a dismal failure. Yet less than three months after they first stepped aboard, the ship experts of the Navy Department had sent her on a trial trip to Cuba and had returned her —in the hands of a green crew—at a speed of 23¾ knots an hour. And this without great pressure upon her engines.

(From "The Sun," Wednesday, May 27, 1914.)

GIANT VATERLAND, IN SAILING, SINKS BARGE.

Finds Hudson Too Small in Pulling Out in First Return Voyage—Almost Backs into Pier—Upheaval Started by Mighty Propellers—Tears Steamships from Lashings.

The pilot of the Hamburg-American colossus Vaterland probably did not meditate on the relation of the length of the ship to the width of the Hudson between the Hamburg-American pier at Hoboken and the piers of the Southern-Pacific Fleet on the Manhattan side of the stream, at the foot of West 11th Street, when he took her out in the fairway yesterday and finally headed her on her first trip to Hamburg.

He might have reflected that less than 6 lengths of Vaterland stretched across the Hudson at that point would have enabled folks to walk the distance dry shod. Also, he might have considered the ponderous displacement of the mighty vessel and the difficulty in checking her momentum under sternway.

Whatever his sentiments may be on the subject, he and Commodore Ruser are the only persons perhaps who knew exactly why the mammoth ship almost backed into Manhattan Island after getting away from her pier yesterday morning, and created a river upheaval in the slips of the Southern Pacific Fleet that was likened to a tidal wave by rivermen who happened to see it. This tore the tankship Topila and the freighter El Valle from the lashings and sunk a coal barge.

WITHIN FOUR FEET OF PIER.

The German ensign floating over the taffrail was within 4 feet of the pier end when the ship began to make headway. It was the narrowest shave that ever Commodore Ruser came to a real accident in all his lucky career. It took only a few minutes for the swiftly backing Vaterland to make Manhattan from Hoboken. The churning of her propellers in going astern sunk in a mist on the usually quiet Hudson, and naturally the vessels in the slips were drawn out to the vortex. Hawsers snapped on a half dozen craft. It looked for a moment as if all would be dragged into the stream. Just then the talent on the liner discovered the proximity of Manhattan and propellers whirled with much more force the other way and a powerful freshet-like current was shot towards the Manhattan shore. The big and little ships that had been sucked out into the channel were sent back on the crest of the combers created by the sudden and swift reversal of the ship's ponderous screws. The force of the wave crest was so great that they leaped over pierheads and spurted in front of bulkheads like breakers on a rocky New England coast. The man who most appreciated the power of the propellers of the greatest liner in the world was the skipper of the coal barge Ulster, of the Dexter & Carpenter Coal Company, which was hurled against the pier and sent to the bottom. He jumped overboard and swam about in the combers until a line was cast to him and he was hauled to safety. A Pennsylvania lighter and the Southern-Pacific liner Oakland were damaged and the piers and bulkheads were torn and dented. The damage which will be paid by the Hamburg-American Line may exceed $10,000.

VICE-DIRECTORS DENY COLLISION.

Vice-Directors J. P. Meyer and W. G. Sickel gave out a statement last evening to correct exaggerated statements about the incident. It said in part: "The damage done in the Southern-Pacific slip was not caused by any collision with the steamship Vaterland, as she never touched any boats, or the pier. It was caused entirely by the swash in turning the ship about. Outside of the sunken coal barge the damage will be covered by a few thousand dollars. No official statement can be made as to the cause of the vessel coming so close to the New York side while turning until an investigation, upon her arrival in Hamburg, can be held. The statement that the vessel left her pier at high speed is not correct. The usual precautions when leaving were observed. It seems unfortunate that any suggestions should be made regarding a speed contest between the Vaterland and the Mauretania. The very emphatic denial by our company, made recently, defines our position."

There was a throng at the Hamburg-American Line pier to see the Vaterland start on her first outward journey, and the heat, greater under the lee of the Hudson hills than elsewhere, caused two women to faint.

Before the lines were cast off Commodore Ruser said that there would probably be no strike of the stewards. It was understood that they had had a conference with the commodore and that he had told them he would do what he could for them after the arrival of the liner in Hamburg. The firemen, it was understood, had similar assurances.

The Vaterland has 515 first cabin passengers, 302 second cabin, and more than 1,000 in her third cabin and steerage.

Mr. E. H. Peabody, *Member*:—There should be some expression of the membership of the Society in regard to this very interesting paper. It is historic, and Mr. Anderson has done us a great service in giving us a paper like this.

Mr. Sperry:—How much was left of the machinery in working order? Could they have gotten home with the machinery as you found it? The forward engines were sufficient with which to send her to sea?

Mr. Anderson:—In reply to Mr. Sperry's question in regard to just how much of the propelling machinery was left in working order, I have pleasure in saying that the ahead turbines were found intact, and leaving out of consideration any question of using the astern turbines, the ahead turbines were perfectly capable of developing full power, namely, 80,000 to 90,000 shaft horse-power.

We did not open out the large low-pressure ahead turbines, but we were able to go inside and thoroughly examine the blading, and this was found to be in capital working order.

We were very anxious to have the turbine rotors turned with the turning gear, but this gear was found to be badly broken, and it took some time before repairs were made, which allowed us to move the mid-pressure rotor, and on July 10 we had the satisfaction of seeing this turbine revolve freely in both directions. In the case of the low-pressure turbines, we were not able to move these rotors until the middle of October, because the turning gear is connected to the line shafting at the aft end of the low-pressure astern turbines, and these engines were being overhauled and repaired.

Mr. Sperry:—I thank you—that is very interesting.

The President:—Then it is your opinion that, as a matter of fact, the machinery was damaged through faulty installation and faulty operation?

Mr. Anderson:—I call it faulty personnel.

The President:—Not through malicious action?

Mr. Anderson:—I am confident the engineers were in very serious trouble before this ship laid up in Hoboken.

The President:—Gentlemen, it needs no comment from me to emphasize the very great importance of the paper presented to us by Mr. Anderson, or to call your attention to the interest which it has created. As a matter of fact, from an engineering standpoint, this paper is one of the important human documents of the engineering side of the war, because on the very rapid recommissioning of that ship and German ships similarly damaged depended to a very great degree the rapidity with which we could send troops abroad. The rapidity with which repairs were made to that ship and other ships in somewhat similar condition had a most depressing influence on the morale of the enemy, and no man can deny truthfully that the quick rehabilitation of those ships was a definite and great factor in the termination of the war.

Therefore, I know that the Society will be very glad to express its special appreciation of the paper presented by Mr. Anderson, and of the comments which have been made by him and other gentlemen in the course of the discussion.

We will now proceed to the next paper on our program entitled, "Standard Lubricating Oil System for Geared Turbines," by Mr. J. Emile Schmeltzer, Member, and Mr. B. G. Fernald, Member.

Mr. Fernald presented the paper.

STANDARD LUBRICATING OIL SYSTEM FOR GEARED TURBINES.

By J. Emile Schmeltzer, Esq., Member, and B. G. Fernald, Esq., Member.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

Prior to July 1, 1918, vessels propelled by geared turbines which had been designed and constructed for private owners had been delivered under requisition to the United States Shipping Board Emergency Fleet Corporation.

About this time an extensive new construction program was planned, which included many contracts for constructing vessels substantially identical with the type which the contractors had been building for private interests. It was not considered necessary, and would have been most detrimental to rapidity of construction, to have started with a clean slate and let the new contracts on completely detailed specifications. It was believed, however, that as experience was gained with machinery of relatively new type, a policy of elimination and standardization both could and should be carried out on the new ships.

As operating troubles with geared propelling turbines were already being reported to an extent considered abnormal, and as many of these troubles were being attributed to faulty operation or failure of the lubricating oil system (in most instances the oiling system was not furnished by the builder of the turbine, but by the shipbuilder), it was decided to study all the oiling systems in use on American ships, and either to adopt or develop a standard oiling system for use on all geared turbine vessels then under construction or to be built by the U. S. Shipping Board Emergency Fleet Corporation.

A very small percentage of American marine engineers had had any previous operating experience with direct-connected marine turbines, and a negligible number had ever operated double reduction geared turbines, the type with which most of the vessels under construction were equipped. This fact, together with the magnitude of the projected shipbuilding program, made it necessary to train many new engineers for marine service, and at the same time indicated the necessity of standardizing, as far as possible, the propelling equipment that they would operate. The standard oiling system was therefore primarily expected to accomplish improvement in operation rather than a reduction in construction costs.

It appeared that a standardized system would materially simplify the instruction of the new engineers. Lack of standardization, moreover, would have caused accidents and confusion, even with experienced engineers, as it was frequently necessary for engineering crews to go to sea on new vessels without first giving them time to familiarize themselves with the different piping systems on the vessels.

It was also believed necessary to incorporate into the system features designed to take care of the unusual war conditions.

A preliminary study of the systems being used by the various shipbuilders disclosed the fact that, with one exception, they were of the direct-pressure type, which, from necessity of keeping all vital parts below the protective deck, had been standardized by the Navy Department. The one exception was a low-pressure gravity system not adapted to turbines requiring high pressure. The fundamental problem was to supply the bearings and gear tooth contact faces with an adequate and uninterrupted supply of clean, cool oil.

This same general problem has been studied for centuries in connection with the water supply for cities and towns, and has been effectively solved, so that, as a basis for a lubricating oil system, the modern city water works was adopted in all of its fundamentals; the main features of which are:—

(*a*) Pumping station.

(*b*) Strainers.

(*c*) Elevated storage reservoir for gravity supply.

(*d*) Filtering system.

(*e*) Delivering and distributing piping.

(*f*) Connection to sewage system.

(*g*) Cooling system. While no artificial cooling system is customarily supplied by a water works, the pipes are carried underground and maintain the original low temperature of the water.

(*h*) Direct pressure emergency system for fire.

In July, 1918, a preliminary plan and description of an oiling system, modeled as above, was prepared and sent broadcast to shipbuilders, turbine builders, lubrication experts, etc., for criticism and suggestions. (See Plan No. E-11000-3, Plate 105.) It was felt that the fundamental plan, being based on a system as reliable as a modern city water works, was beyond serious criticism; but that its application to shipboard conditions may have been faulty in detail. Much constructive criticism was received and much that was essentially a defense or justification of systems to which the critic was already committed by precedent, or in which he had some proprietary interest.

We are particularly indebted for the cooperation of Mr. Chas. F. Bailey, engineering director of the Newport News Shipbuilding and Dry Dock Co., for his constructive criticism and instructive suggestions, based not only on his extensive general marine experience but on a specialized study of the faults of existing systems disclosed at the time many of these vessels called at the yards of his company for emergency repairs.

We are also indebted to Mr. Tobin, of the Vacuum Oil Company, for data regarding lubricating oils and their actions under various conditions met with in service.

As a consequence of the criticisms, a revision of the system substantially according with the majority of the recommendations was made and submitted again to a number of eminent marine engineers, and the final revised system adopted as a standard for vessels of the Emergency Fleet Corporation on September 16, 1918. (See Plan No. E-11000-1, Plate 106.)

The actual installation or application of the system was materially delayed by the necessity for speeding up our entire war program after the German spring drive. Materials were being ordered and ships constructed by methods which would not allow of the simplest change in construction without a delay too serious to be allowed under the emergency. Certain essential improvements in existing systems were, however, made in advance of the adoption of the entire system, *e. g.*, oil coolers and pumps of more liberal capacity were furnished, and vital parts of the system were installed in duplicate instead of singly.

Late in the summer of 1918, it was decided to install the system in accordance with Plan No. E-11000-3 (Plate 105), on the S. S. Westland, then in Hoboken undergoing repairs. Due to the short space of time allowed, it was impossible to make the installation as complete as desired, but the necessary information as to the pumps, coolers and other equipment required was furnished to the Division of Operations, and they were installed under the direction of the representative of that division in New York. While the system as specified in Technical Order No. 75 contains many refinements not included in the system installed on the Westland, the results obtained from the installation on this vessel were sufficient to assure those responsible for its design of the successful operation of the proposed system. The first vessels equipped with the entire system were delivered about the middle of January, 1919, subsequent to the signing of the armistice.

As the shipbuilding program, notwithstanding many cancellations, was far from complete, it was deemed advisable to make a further revision of the system to eliminate many features specified on account of war conditions, and also to eliminate automatic features rendered unnecessary by the increasing efficiency of the operating personnel, but retaining the essential features.

This revision was issued on July 15, 1919, and the departures from the system previously issued are noted therein. (See Technical Order No. 75 and enclosures, Plan No. E-11000-10, Plate 108.)

Coolers.—Nearly all of the oil coolers previously used were inadequate to transfer the required number of British thermal units, which amounted in a number of instances to 10 per cent of the propelling turbine horse-power. This was partly due to the fact that the reduction-gear manufacturers had overrated the efficiency of their reduction gears.

As a precautionary measure, it was considered essential that an auxiliary cooler be provided so that the oil could be cooled by one cooler in the event of the other being rendered inoperative by leaky tubes or other causes. The auxiliary cooler was also used in parallel with the main cooler when operating in tropical waters, where a cooling water temperature of 84° or higher obtained. The technical order specified a 20° drop in oil temperature, and, while it was realized that the cooler would not be called upon to equal this temperature drop under normal running conditions, still this size of cooler provided for a considerable drop in the efficiency of the cooler, due to the fact that the tubes would in time become covered with a deposit which would materially decrease the efficiency of the cooler as compared with a clean one.

Results obtained in operation have demonstrated that the size of cooler specified satisfies average operating conditions.

Gravity Tanks—Size and Location of.—During the time the designs of the new system were in the course of preparation, one manufacturer claimed that it was necessary to have 12 pounds oil pressure on the turbine bearings and gear oiling devices on equipment manufactured by them, and still another used spray nozzles of such size as to make it almost impossible to maintain a higher pressure than 4 or 5 pounds. The rest of the builders' requirements averaged about 10 pounds pressure at the oil manifold.

Due to the above facts, and based on the assumption that it was best to satisfy the higher pressure requirements, and also owing to the fact that all of the vessels on which installation of the system was contemplated would permit of the location, it was specified that the gravity tanks be located at a height of 30 feet above center line of turbine in the engine-room casing. This location of the gravity tank has proved very satisfactory in service, and no trouble has been experienced in securing the required pressure.

The size of the gravity tank, which is of 600 gallons capacity, was based on having three or four minutes reserve supply of oil available for the gears in the event of the oil supply to the gravity tanks being interrupted. This, it was thought, would provide sufficient time for action on the part of the operating personnel.

Kingsbury Thrust Bearing.—Inasmuch as, on most of the various types of turbines purchased by the U. S. Shipping Board, the Kingsbury or similar type of thrust bearing is used, it is essential that there be a constant supply of cool, clean oil. Failure or interruption of the supply has caused the thrust bearing to burn out on several occasions, with consequent serious damage to the turbine.

This same type of thrust bearing was employed in the majority of cases for the main propelling shaft thrust, and therefore the same necessity for cool, clean oil obtained in this case.

Strainers.—As a precautionary measure against sabotage, a twin suction strainer having a coarse mesh strainer basket was installed, but eventually omitted in the revised specifications at the end of the war.

A strainer having approximately 1/64-inch mesh was placed in the discharge line, and this is considered sufficient to remove all sediment incidental to ordinary operation after the system has been thoroughly cleaned before the trial trip as specified.

Filters and Separators.—While all of the filters and separators specified are satisfactory for the purpose, it is considered that the results obtained from the centrifugal type of separator show greater efficiency, inasmuch as the oil, water and sediment are separated—the oil flowing from one nozzle, the water from another, the overflow mixture from another, and the sediment remaining in the bowl, from which it may be removed at will. Tests made at Annapolis prove that the rolling or pitching of a vessel will have no ill effects on its successful operation.

Filtering elements placed in gravity tanks or in the discharge line from pumps to gravity tanks at its connection to gravity tank, unless at sufficient height above, cannot function properly, due to the lack of sufficient head, which results in the oil following the path of least resistance, *i. e.*, through the overflow and not through the straining elements as intended.

Alarm Systems.—An electric alarm system actuated by a float switch, connected to the gravity tanks was provided (and since omitted), which would notify the chief engineer in his room and the engineer officer on watch, by means of an electric gong, that the oil level had dropped below normal operating level. This system, which was so designed as to not only give the necessary notification in case of oil level becoming low but also to automatically increase the speed of the pumps, was later omitted, due to a desire to reduce the cost of the installation rather than because of any belief in a lack of necessity for it.

Drain Tank.—The capacity of the oil drain tank was required to be from 800 to 1,000 gallons for a 3,000 horse-power double-reduction gear turbine unit.

It was found that most of the drain tanks supplied at the time this system was originated were entirely too small, and resulted in the pumps becoming vapor bound. In most cases of this kind, and also where the suction lift of the pump was high, the oil became very dark in color. Oil experts who were consulted advised that the mixture of air with hot oil frequently resulted in the presence of sulphurous acid (H_2SO_3) in the oil. For this reason, the drain tanks were made sufficiently large and the pumps placed as low as possible to insure a short suction lift.

Pumps.—While vertical pumps are more desirable for the purpose of securing low suction lift, it was not possible to procure them in the quantities required. Horizontal pumps were therefore used more extensively. The vertical pump is preferred to the horizontal, not only for the above good reason but also because of its taking less floor space and lending itself to better arrangement generally.

Oils.—Originally an oil having a viscosity of 300 seconds (Saybolt) at 100° F. was specified, but later this was changed to 500 seconds viscosity. This latter viscosity was found to suit conditions better because the high-speed bearings demand oil having a low viscosity and the gears and slow-speed bearings a high viscosity oil. Inasmuch as two separate systems could not be considered on account of their cost, and due to the fact that oils ranging from 300 to 700 viscosity at 100° have practically the same viscosity at temperatures above 140°, and also based on operating results, it was considered that the higher viscosity was more desirable.

Conclusions.—The results obtained from the system in service have been very gratifying and show that the sizes of equipment specified were in keeping with the requirements, and it is believed that a system of lesser magnitude than that specified would endanger the successful operation of a double reduction gear turbine unit of 3,000 horse-power.

There is no doubt that an adequate lubricating oil system is essential to the successful operation of the geared turbine, and while its maintenance cost is small, the consumption of oil through leaks and other sources amounting to about half a barrel per month, the first cost of installing such a system is quite high. It is believed, however, that an increase in efficiency will be obtained in the reduction gears of the future, which will permit of a considerable reduction in the size, and consequently the cost of the lubricating oil system.

List of Documents and Plates.—(*a*) Plan E-11000-3, showing proposed arrangement of lubricating oil system for units up to and including 3,000 horse-power, dated July 26, 1918. (See Plate 105.)

(*b*) Plan E-11000-1, showing arrangement of lubricating oil system for geared turbine units up to and including 3,000 horse-power, dated September 9, 1918. (See Plate 106.)

(*c*) Technical Order No. 75 with documents as amended July 15, 1919. (See pages 248 to 262, and Plates 107, 108 and 109.)

(*d*) Specifications for electric alarm system. (See pages 262 to 264.)

(*e*) Photograph of discharge strainer. (Plate 110.)

(*f*) Photograph of discharge strainer, disassembled. (Plate 111.)

(*g*) Photograph of centrifugal separator. (Plate 112.)

(*h*) Sectional cut of centrifugal separator. (Plate 113.)

(*j*) Photograph of filter. (Plate 114.)

(*k*) Arrangement plan of electric alarm system. (Plate 115.)

DOCUMENT C.

UNITED STATES SHIPPING BOARD, EMERGENCY FLEET CORPORATION, PHILADELPHIA, PA.

CONSTRUCTION DIVISION.

Technical Order No. 75.

July 15, 1919 (9/16/18). (As amended) 7/15/19.

To: District Managers and Turbine Inspectors for Action.

To: Steel Shipbuilders and Others for Information.

Subject: Standard Lubricating Oil System for Geared Turbines.

Enclosures: (*a*) One copy of specifications No. K-275-T—3,000 for lubricating oil system for geared-turbine units of approximately 3,000 horse-power, and application of system to other sizes, and twin-screw installations.

(*b*) List of approved equipment.

(*c*) Copy of Navy Department Print No. B-56, standard thermometer fittings. (Plate 107.)

(*d*) Arrangement of lubricating oil system for geared turbine units up to and including 3,000 horse-power. Drawing No. E-11000-10. (Plate 108.)

(*e*) Temperature—Viscosity curves for approved lubricating oils. Drawing E-11000-11. (Plate 109.)

A. General.—1. Technical Order No. 75, as issued on September 16, 1918, and its amendment of November 8, 1918, provided for certain features which were necessary precautionary measures while the country was at war, and which the attached specifications and other enclosures will eliminate. It has also been found that there has been a tendency in some quarters to follow the diagrammatic plan, document *d*, as though it were a working drawing, and not merely an outline, as it was intended.

2. Due to the urgency of the situation, and for the sake of brevity, little space was devoted in the first issue of Technical Order No. 75, and its amendment of November 8, to explanations or reasons why certain provisions were made. It is believed that the documents hereto attached will not only set forth requirements clearly but will satisfactorily explain the reasons for their necessity.

B. Departure from Previous Order.—The departures in this amendment from Technical Order No. 75, and its amendment of November 8, are in general as follows:—

1. Suction strainer has been omitted. This strainer was originally intended as a precautionary measure against sabotage, and to prevent foreign matter, left in the system through carelessness, from entering pumps, and thereby causing serious damage. The ending of the war, and the general adoption of an effective method of cleaning the system, before trials, have rendered the use of this strainer unnecessary.

2. Two reserve tanks have been omitted. Extra reserve tanks were required so that, in the event of the two gravity tanks being rendered useless by gun fire, the reserve tanks could be used as gravity tanks. Another reason was the necessity for carrying a large oil reserve due to the impossibility of securing suitable oil in foreign ports.

3. By-pass in oil line for the purpose of by-passing gravity tanks, and pumping direct to supply line, has been omitted. This connection was provided so that, in the event of gravity tanks being shot away, oil could be pumped directly from drain tank to system, but it is no longer necessary.

4. Gauge type indicator with electric contacts in discharge line, between pumps and discharge strainer, has been omitted. With increasing familiarity of the operating force with the system, this additional safeguard is now believed unnecessary. Ordinary pressure gauges reading from 0 to 100 pounds should be installed in their place. Float switches in gravity tanks and accompanying electrical arm system have also been omitted.

5. The overflow from top of gravity tanks has been changed to side of tanks as near the top as possible. Experience with systems installed with overflow discharging from the top of tank proved that the air was not satisfactorily released through the vent, due to lack of space at the top of tank for this purpose. The new arrangement has been found to work satisfactorily in service.

6. Sight flows and direction indicators of any kind have been eliminated from supply line inasmuch as a practical device is not generally procurable.

7. Oils having a minimum viscosity of 500 seconds (Saybolt) at 100° F. have been substituted for the previous ones. This decision was reached as a result of our experience to date and by agreement with the Division of Operations.

8. Thermometer fittings have been changed from S. E. print B-58 to B-56 which shows a shorter separable socket.

9. Various changes of greater or less importance have been made throughout, and it is suggested that the various documents be compared with those previously issued, with the end in view of incorporating all of the changes in future installations.

C. Application of Present Order.—1. Plan E-11000-10 shows an arrangement of the lubricating oil system for geared turbine units up to and including 3,000 shaft horse-power, and while the plan is purely diagrammatic, there is no doubt but that it can be readily adapted to any of the various types of turbine-driven ships.

2. For vessels in an advanced stage of construction, the change from the system called for in the original Technical Order No. 75, to the arrangement specified in the foregoing, is to be carried into effect only where it will not seriously delay the completion of the vessels, or where it can be effected at a reasonable cost. If serious delay or excessive cost is anticipated, the matter shall be referred to the Home Office before final action is taken.

3. Where piping, manifolds, fittings and other material is already available or provided for, it shall be utilized and rearranged to conform as closely as possible to the above requirements, in order to minimize cost and avoid waste and scrapping of material.

4. Where the conditions stated in paragraph 2 do not obtain, the revised requirements should be made effective on receipt.

D. Adjustment of Changes in Cost.—1. Where any of the equipment herein specified is not required by the contract specifications, or by subsequent agreements, it will be furnished and installed as a change under the contract, and the matter of cost shall be dealt with in the usual manner.

2. As, however, the departures from the original system consist mainly in omissions and simplifications, a substantial decrease in cost is anticipated.

3. On vessels in which the change to the present requirements can be effected, and for which the cost has already been adjusted on the basis of the original system, the matter of decreased cost shall be handled in the usual manner.

E. Agreement with Division of Operations.—The revised system specified herein conforms with the views of the Division of Operations, Department of Construction and Repair.

F. Approval of Drawings.—General arrangement and detail drawings of oiling system must be submitted, with district manager's comments, to the Engineering Section before final approval.

(Signed) P. J. McAuliffe,
Manager, Ship Construction Division.

Documents (5).

UNITED STATES SHIPPING BOARD, EMERGENCY FLEET CORPORATION, PHILADELPHIA, PA.

DOCUMENT A.

Technical Order No. 75 as Amended, 7/15/19.

Detail Specification No. K-275 T. 3,000.

Lubricating Oil System for Geared Turbine Units up to and including 3,000 horse-power.

A. General Description.—The system generally shall consist of a drain or sump tank; a duplex lubricating oil pump, and an auxiliary pump of the same size; two oil coolers; two gravity tanks, either of which may be used as a settling tank; one twin oil discharge strainer; an oil filter or separator; a cooling water-circulating pump; one reserve oil tank; piping and connections; and shall conform to these specifications and to Drawing No. E-11000-10. (Plate 108.)

1. The oil will be drawn from the drain or sump tank by either main or auxiliary oil pump, and discharged through a twin oil strainer to either or both oil coolers, and thence to gravity tanks. From these tanks the oil will discharge through an internal pipe (extending 8 inches above the bottom of the gravity tank) to a line leading to all bearings and gears, thence drained to sump tank.

2. A 1½-inch drain line shall be led from the lowest point of the gravity and reserve tanks through an oil filter or separator to the drain tank, so that oil may be filtered continuously or at stated periods.

3. An overflow line shall run from the side, and as near the top as possible, of the gravity tanks to the drain tank. This gives the air a good chance to be released from the oil through the vent pipes.

4. A connection on oil discharge line will allow the oil to be discharged direct to bilge, in the event of oil becoming emulsified. Control of this line shall be by means of suitable cock with name plate clearly marked as follows:—"For emergency use only to discharge badly emulsified oil to bilge. Turbine to be stopped before using."

5. Piping shall be arranged to permit of by-passing coolers so that oil may be discharged directly from pump to gravity tank. This has been found necessary when using heavy oils with cooling water of low temperature.

6. Piping shall be arranged so as to permit either gravity tank being used alternately as a settling tank, in which oil shall be allowed to settle and then be drained to filter or separator. Under normal operating conditions either gravity tank shall constantly overflow to drain tank.

7. Piping shall be arranged to permit of oil being delivered from reserve oil tank to the filter or separator, and to drain tank through overflow line from gravity tanks.

8. Pressure gauges shall be provided in accessible and visible positions in discharge lines from both oil and water pumps and in oil supply manifolds to turbine

and gears, and also at oil cooler outlets. The gauges showing the pressure of oil from the cooler and the water pressure into the cooler should be placed at the same level, and the oil pressure gauge should always read higher than the water gauge. If there is a leak, it is preferable that it shall be from the oil to the water.

9. Filling line to gravity and reserve tanks shall lead from a convenient location on deck, through a swinging pipe under the deck, to highest point of discharge line to funnel and valve, as shown in drawing E-11000-10 (Plate 108.) All tanks shall be properly vented with large goose-neck vent pipes extending at least 3 feet high. The vent pipe on gravity tanks shall be the same size as overflow pipe, and may be connected by tee fitting to overflow pipe where it connects to gravity tanks. Oil drain piping from bottom of gravity tanks shall be led directly with uniform drop through filter to drain tank.

10. All pipe sizes shall be determined on the basis of 175 gallons of oil per minute (the oil having a viscosity of 500 seconds Saybolt at 100° F.) at the following velocities:—Oil in drain return pipes from bearings and gears not to exceed 80 feet per minute; oil in suction lines not to exceed 125 feet per minute; oil in discharge and supply lines not to exceed 250 feet per minute. Due allowance shall be made for friction in bends and fittings, in order that pressure of oil at supply manifolds shall be at least 10 pounds per square inch.

11. There shall be a glass sight flow indicator installed in the overflow line fitted with electric light, and so located as to be visible from operating stations. There shall also be connected to the gravity tanks three ⅜-inch lines at high, intermediate and low levels of oil in the gravity tanks respectively and these pipes will be led through sight flows on engine-room gauge board to the drain tank. The glass of indicator shall be marked as follows:—"High Level"—"Intermediate Level"—"Low Level." When oil stops flowing at "Low Level," shut down turbines and investigate, an electric light to be placed behind indicator.

Approved type of oil flow indicator or telltales with vapor or air relief shall be mounted on each bearing.

12. Where cocks are installed in supply lines to bearings or oil sprays, for initial adjustment of flow, they shall be of a type requiring a special handle or key (kept in chief engineer's room). They shall also be arranged to permit a small amount of oil to flow even if left in "shut" position. Only cocks will be allowed for this control, as there is a danger of valves closing in the event of heavy vibrations.

B. Thermometers.—Thermometers shall be 9 inches, separate socket, Class A as per Navy Department S. E. Print B-56, graduated from 30° F. to 220° F., located as indicated on Plan E-11000-10 (Plate 108), *i. e.,* one at turbine thrust discharge and one at forward turbine bearing oil outlet; one at after turbine bearing oil outlet; one at combined oil drain from gear casing; one at main thrust bearing oil outlet; one at oil inlet to coolers; one at oil outlet from coolers; one at cooling water inlet, and one at cooling water outlet. A thermometer shall also be placed on each gravity tank. If cross-compound turbines are used, the thermometers above specified will be required on each turbine. Thermometers in turbine bearings must be protected from gland leak-off steam, so as to avoid false readings.

C. Gravity Tanks.—1. Two cylindrical gravity or settling tanks shall be located either side of center line of ship at top of engine-room casing. The bottom of tanks shall be 30 feet above center line of turbines.

2. Each tank shall have a capacity of 600 gallons of lubricating oil, and shall be provided with heating coils for heating oil to 180° F., preparatory to filtering.

3. Manholes shall be provided to permit accessibility for cleaning.

4. Gauge glasses about 20 inches long shall be provided at top of both tanks, for observation purposes when filling.

5. The drain line from gravity tanks shall have a connection, with valves, to an open funnel leading by separate line direct to bilge so that water and sludge may be drained to bilge after settling either tank preparatory to filtering.

6. An air vent shall be connected to top of each gravity tank as shown on Drawing E-11000-10 (Plate 108).

D. Reserve or Make-up Tank.—1. One reserve tank of 600 gallons capacity shall be conveniently located as high as possible below main deck in engine-room connected to overflow line leading to drain tank by a pipe with internal connection carried 8 inches above lowest point of tank.

2. Gauge glasses about 20 inches long and in series, with ends overlapping, shall be provided for reserve tank.

3. Manholes shall be provided to permit accessibility for cleaning.

4. A vent similar to that on gravity tanks shall be fitted and 1½ inches drain connection to filter shall be taken from bottom of tank.

E. Drain or Sump Tank.—1. Drain tank shall be separate from ship's structure and located as near as possible to center line of ship, and shall always be located abaft of the gear case. It shall be of sufficient capacity above top of internal suction fitting to hold the oil of one gravity tank plus oil in supply line. Should it be impossible to fit a single tank of required capacity, then two tanks may be fitted with cross-connection and separate vents.

2. Drain tank shall be provided with two vertical baffle plates equally spaced, which shall extend from within 1 inch of bottom to about one-third of total depth from drain tank top.

3. Float shall be provided in drain tank, and shall have indicator fixed to extension on vertical stem leading from float which shall indicate oil level by graduations engraved on pipe or casing enclosing stem, and indicator and piping shall be so arranged that it is easily visible at all times, and at such a height that tank cannot overflow through indicator pipe when vessel is in a seaway.

4. The suction connection from pumps to drain tank shall be by means of an internal elbow having a bell mouth and extending to within 2 inches of the bottom of the tank. The suction shall be taken from the center of after end of the tank, so that any trim of the ship will increase the submersion of suction pipe.

5. A manhole shall be provided for entering tank and cleaning it, and in the

vertical baffle plates lightening holes of manhole size shall be cut to clean out all compartments, and so staggered as not to interfere with baffling effect.

6. Drain pipe from bearings and gear case shall be led with an easy drop to the top, or as near the top of the drain tank as possible; oil drains from turbine bearings must not be led into the base of the gear case.

7. A hand pump for the purpose of sampling oil from the bottom of drain tank shall be provided.

8. Drain tank to be vented by goose-neck pipe extending at least to a height equal in elevation to center line of turbine.

9. Drain tank shall be fitted with small cock as near bottom as possible, to drain off bottom oil to bilge, when tank is to be cleaned out, and it shall have handle kept in chief engineer's room. Cock shall be marked with plate as follows:—"This cock only to be used when cleaning drain tank and cock to be shut and handle returned to chief engineer's room when cleaning is finished."

F. Lubricating Oil Pumps.—1. A duplex lubricating oil pump and an auxiliary pump of the same size shall be placed in such a location as to permit of a suction head not exceeding 2 feet. They shall conform to the following specifications:—

Size 7½ by 7 by 10 inches duplex, preferably of the vertical type; pumps of other dimensions, but with equivalent capacity, may be used. Pumps shall be designed to operate from the auxiliary steam line at reduced pressure, but the steam ends must be tested hydrostatically for full boiler pressure.

Pumps shall have:—Brass valves, cast-iron plunger, and steel pump rods. Cast-iron snap rings on piston and plunger. Steam and oil piston rods packed with metallic packing. Steam cylinders lagged, and covered with sheet metal neatly fitted. Paper packing on all joints. No liner in the oil end of the pump. Both suction and discharge connections of the flange type, and the dimension shall conform with the American standard.

G. Oil Coolers.—1. Two oil coolers shall be provided, each capable of cooling 175 gallons per minute of lubricating oil (having a viscosity of 500 seconds Saybolt at 100° F.) from 130° F. to 110° F., with cooling water at 70° F. The drop in oil pressure through the cooler shall not exceed 15 pounds per square inch. The drop in water pressure through the cooler shall not exceed 5 pounds per square inch.

2. Coolers shall be located below light load water line if possible.

3. Piping arrangement shall permit of by-passing from one cooler to the other, or from both coolers direct to gravity tanks.

4. At the lowest point of the water head on each cooler there shall be placed a drain discharging into an open funnel. When ship is in port, this drain shall be opened so that all of the water will be removed from the cooler and discharged through a funnel to the bilge. By leaving this drain open while in port, any leak or defective tube will be shown by oil dripping into the funnel.

5. Cooling water shall be supplied by a separate circulating pump with an emergency connection to an intermittent service pump other than the sanitary pump, having a direct connection to sea, but means must be provided to insure through

the use of a relief valve that water pressure at cooler inlet should always be less than the outlet pressure of oil from the cooler. As this requirement depends on the arrangement adopted and may necessitate a low setting relief valve on circulating water inlet, with consequent possibility of continuous overflowing to bilges, a spring loaded valve must be placed on oil discharge line from cooler, thus enabling the oil pressure to be artificially raised by adjusting this valve.

6. The pump shall be of the following specifications:—

Size 7½ by 7 by 10 duplex; pump of other dimensions but with equivalent capacity may be used. Pump shall be designed to operate from the auxiliary steam line at reduced pressure, but the steam ends must be tested hydrostatically for full boiler pressure.

Pump shall have:—Brass valves, bronze plunger and pump rods. Cast-iron snap rings in pistons. Tufts or equal packing in plunger. Brass liner in water end of pumps. Metallic packing for steam piston rods. Steam cylinder lagged, and covered with sheet metal neatly fitted. Both suction and discharge connections of the flange type, and the dimension shall conform with the American standard.

7. Oil cooler water circulating pump shall discharge through cooler overboard.

H. Strainers, Filter or Separators.—1. A strainer of the twin type shall be located in the pump discharge line adjacent to the pumps, and in such a position as to permit the removal of the strainer baskets by lifting them upward. Care should be taken that strainers are located in such a manner as to prevent the sediment falling back into the strainer upon removal of the baskets, or loss of oil when strainer is cleaned.

2. Strainers are to be so designed that one side can be cleaned while the other is in operation. The valve stems shall be so interlocked that when one valve is closed the other is open. Two spare baskets must be supplied, also a bucket, and a supply of kerosene for cleaning. Strainer castings shall be pickled and thoroughly cleaned before shipment. Internal surfaces in contact with the oil shall not be painted or otherwise coated. Baskets shall be of copper perforated with holes .020 inch in diameter. The ratio of straining area to inlet area shall be at least 6 to 1. A plate containing adequate instructions for operation shall be placed on each strainer.

3. Either a centrifugal separator, or a gravity filter of approved type, must be installed as indicated on Drawing E-11000-10 (Plate 108), with connection for oil supply to separator or filter through both gravity tanks and the reserve tank. The other connections required by the instruction sheet accompanying the separator or filter shall also be made. A centrifugal separator will be found especially effective in breaking up an emulsion of water and oil, even in a rough sea. The centrifugal separator can be either electric motor or steam driven. If turbine drive is used the design should provide for operation against 10 pounds back pressure, so that the turbine can exhaust into the heater. The gravity filter, while effective in cleaning the oil, requires the heating and settling to break up an emulsion. Its effective-

ness depends on the use of clean filter cloths. An ample supply of special lintless cloth should be carried on board, and frequently changed.

I. Piping.—1. All piping from discharge side of manifold to bearings and gears shall be either copper or brass. The drain and suction piping shall be either lap-welded, steel or seamless drawn steel.

2. All screwed piping to be reamed at ends to remove burrs.

3. Flanged joints shall be made to all tanks, strainers, coolers, filter or separator, pumps, turbines, gear casings and manifolds, and in all cases in brass piping connections. Flanges shall be brazed to brass or copper pipes, Screwed flanges may be used with steel pipe.

4. Sufficient flanged joints and unions to be provided to facilitate inspection and dismantling of piping.

5. All iron or steel pipes, fittings, tanks and other accessories forming a part of the lubricating oil system shall be thoroughly cleaned by pickling and sand-blasting before being installed. Cored holes in oil manifolds or castings shall be bored or reamed out to insure clean surfaces.

6. A cock shall be fitted at gear casing discharge. This cock shall be so constructed that it will lock open by means of a handle kept in the chief engineer's room. A plate shall be fitted to the gear case above the cock, reading as follows:—"Cock at bottom of gear case shall be used only when it is desired to clean out drain tank. This cock is then to be closed, and bottom half of gear case used as a temporary tank for clean oil from gravity tank. Care must be taken to see that oil does not rise above level of test cock, which is a safe distance below bottom of journal. Handle shall be returned to the chief engineer's room after cock has been locked open again."

7. A test cock shall be fitted to gear case 6 inches below the lowest point of any gear or pinion shaft, so that when gear case is being used for temporary storage as outlined above, it shall not be filled high enough to flood bearings with oil. The cock shall be marked "For use when utilizing bottom half of gear case as temporary tank. When oil appears, valve at gravity tank to be immediately closed to prevent oil flooding bearings."

8. After installation, lubricating oil shall be heated to 180° F. and pumped through system, and carefully strained for a period of twenty-four hours (previous to running turbines) with upper bearing shell removed and bearing caps replaced.

9. A locked valve with hose connection is to be placed on oil discharge line at a point near main deck to facilitate emptying system into barrels if necessary. Suitable length of oil hose to be supplied.

J. Operation of System.—1. *Filling*: When putting fresh oil into system from barrel or drum through deck plug, care must be taken that same is done through a funnel which screws into deck plug, and has a suitable wire mesh screen in it. This funnel is to be supplied by shipbuilder.

2. *Condition of system when fully charged with oil.*—Reserve tank full and shut off from system. One gravity tank full and shut off from system. System, in-

cluding pumps, strainer, coolers and piping full and level of oil in drain tank just above top of horizontal pipe connecting to internal suction fitting, and other gravity tank full to overflow pipe.

3. *Care of strainers.*—When the strainer in use has to be cleaned, and it is necessary to change over, the dirty basket shall be removed and replaced by a clean spare immediately, or if no spare is available the dirty basket shall be thoroughly cleaned and replaced, and in either case the cover shall be left open until the replacement is made. This precaution will reduce the liability of dirty oil being supplied to the turbines and gears by a quick change over to the side of the strainer from which the basket has been removed.

4. *Rurifying when running.*—This can be accomplished by observing the following procedure:—

(*a*) Open valve on supply line to turbines and gears on gravity tank which contains purified oil.

(*b*) Shut valve on supply line to turbines and gears on gravity tank which has dirty oil and has just been in use.

(*c*) Open valve on pump discharge line to gravity tank which has just been cut in.

(*d*) Close valve on pump discharge line to gravity tank which has just been cut out.

(*e*) Turn steam on dirty oil tank and heat oil to a temperature of 180° F. for twelve hours. Steam can now be shut off coils and oil allowed to settle for an additional hour.

(*f*) Close all drain valves on both gravity tanks and the one on drain line to filter.

(*g*) Open valve to open funnel on drain line direct to bilge, and let water and sludge run off until clean oil appears, when both these valves are to be shut. Gravity tank is then ready for cutting in on system when required.

(*h*) Repeat process with other gravity tank when necessary.

5. *Filtration.*—Continuous or batch filtration should be used as much as possible, and should be carried out on gravity tank which is in service and running on the system. It must be clearly understood that this process is really filtering a portion of the working oil in the whole system. Filtering elements should have cloths renewed at suitable periods to be determined by running conditions. The process is as follows:—

(*a*) Open drain to filter line, open drain below tank, and gradually regulate oil so as to slowly flow to filter when continuous filtration can be carried on as per instructions provided by manufacturers of filtering apparatus.

(*b*) If the ship is rolling heavily then batch filtration can be carried on, for which manufacturer's instructions are also provided, depending on type of filter.

(*c*) When tanks are changed over, the other tank will then be filtered as above.

6. *Purifying when vessel is in port and turbines shut down.*—Upon ship's arrival in port, attention should be turned to purifying lubricating oil to the highest

degree possible in readiness for the next voyage. Lubricating system should be allowed to run for half an hour after "Finished Signals" is rung down, to cool down bearings, etc., and then all valves, pumps and tanks should be completely shut down and procedure commenced as follows:—

(*a*) Open test cock on bottom of gear case to indicate when oil is at maximum height allowed without flooding bearings. Key to this cock to be obtained from chief engineer's room.

(*b*) Shut locked cock on main oil drain line at bottom of gear case. Key to be obtained from chief engineer's room.

(*c*) Take up floor plate and remove manhole from drain tank.

(*d*) Slightly open and regulate valve on supply line to turbines and gears of cleanest tank, gradually lower oil into gear case by an amount not to exceed the height of test cock, and when oil appears at this cock, valve on gravity tank must be instantly closed.

(*e*) Open valve on pump discharge line at gravity tank in which oil has just been lowered, and slowly pump up the oil in drain tank into it until drain tank is pumped as low as pump suction will take it.

(*f*) Take out remaining oil in drain tank with hand sampling pump if possible. If any remaining oil is good, it can be removed by hand. After getting key from chief engineer's room, open drain cock on bottom of drain tank and drain as much as possible of the remaining oil to bilge. Shut cock and return key to chief engineer's room, and clean bottom of tank out by means of cloths (not cotton waste) and kerosene until all sediment is out and tank wiped dry, when manhole cover is to be replaced.

(*g*) Open steam on coil to both gravity tanks until oil is shown at temperature of 180° F., letting oil remain at this temperature for twelve hours, when steam should be shut off and oil allowed to settle for an additional hour.

(*h*) With control valve on drain line to filter closed, and the valve on drain line direct to bilge below funnel open, the drain valve on each tank should be opened one at a time, and water and sludge drained to bilge. When pure oil appears in each case, shut off all drain line valves.

(*i*) Open and regulate valve on drain line to filter from one gravity tank, allowing oil either to filter continuously or in batches to drain tank, renewing filter cloths when necessary where centrifugal type is not used. In this manner, completely drain off one gravity tank to drain tank and close drain valve.

(*j*) Open up pump discharge valve at gravity tank just filtered. Start up pump, and pump as much of this oil as suction will allow, back to tank.

(*k*) Repeat above process with second tank, as per paragraphs (*i*) and (*j*).

(*l*) Now open cock on bottom of gear case and allow oil to flow to drain tank.

(*m*) Shut cock again.

(*n*) Open up valve on supply line at bottom of one of gravity tanks, and allow filtered oil to flow through gears until at level of test cock, when valve on gravity tank must be immediately closed.

(*o*) Pump up oil in drain tank to partly filled gravity tank which should again fill it.

(*p*) Open up drain valve at bottom of this gravity tank and allow oil to flow through filter to drain tank and close drain valve again.

(*q*) Pump all oil in drain tank up to empty gravity tank.

(*r*) Open cock on bottom of gear case and allow oil to run into drain tank locking cock open, and returning key to chief engineer's room; close test cock and return its key to chief engineer's room also. This is important.

(*s*) System is now ready to operate with pure oil for another voyage.

K. Limitation of System.—1. After the oil system is in operation and turbines running, it is to be noted that the oil system is not absolutely automatic in the following points:—

(*a*) *Gravity tanks.*—Supply line control valves. Valves controlling gauge board indicator pipes. (All to be operated by hand when changing over from one tank to the other.)

Operating heating coils, drains and water sludge valves when settling the gravity tank with system closed off.

(*b*) *Turbines and gears.*—Regulating supply from gravity tank and valves (if fitted) at turbine manifolds.

(*c*) *Oil pumps.*—Changing from main to auxiliary oil pumps. Regulating speed of pumps by control valves.

(*d*) *Strainer.*—The strainer requires hand operation to change sides so as to use a clean instead of a dirty basket. It also requires immediate replacement of a dirty basket by a clean one, when the former is removed.

(*e*) *Coolers.*—Opening up cooler or coolers on line or by-passing coolers. Changing from one oil cooler to another, by-passing both or putting both into operation.

(*f*) *Filter.*—Operating filter on batch or continuous system or cutting filter in or out. Operating drain or filter and adjusting water overflow, etc. Cleaning filter cloths.

(*g*) *Pipe line.*—Operating valve on pump discharge line controlling discharge of badly emulsified oil from drain tank to bilge.

(*h*) *Filling.*—Operating valves for filling from deck.

(*i*) *Reserve tank.*—Operating valves controlling reserve supply for make up oil.

L. Application of Standard System to Installations Other than 2,500 to 3,000 S. H. P. Single-Screw Units:—

1. *Single-screw installation.*—The standard system is intended for use on all double reduction geared units from 2,500 to 3,000 horse-power in capacity, whether single or compound turbines are used. For single reduction geared units which require less oil, or for sizes smaller or larger than those given above, the size of tanks and the capacity of the system throughout shall be arranged, in the same proportion as the standard system, on the amount of oil per minute required by the unit selected, *e. g.*, a turbine unit requiring a circulation of 87½ gallons of oil per minute

would use tanks, etc., approximately half the size of these specified. All proposed designs must, however, be submitted to the Engineering Section for approval.

2. *Twin-screw installation.*—The standard system will, in general, apply to twin-screw installations, requiring approximately 175 gallons of oil per minute. The only departure from the standard system will consist in branching the supply piping from gravity tanks to each turbine. The return oil piping from turbine bearings and gears may also be led to one drain tank common to both turbines. In twin-screw installations requiring a greater capacity than the standard, each turbine should be arranged with its own oiling system, and the two systems should be cross-connected so that either system may be used on both turbines. Depending on the size of the installation, one spare gravity tank and one spare lubricating pump for the complete system may be used instead of two; but for high power installations it is preferable that each system should be in duplicate. All proposed designs, however, must be submitted to the Engineering Section for approval.

M. Lubricating Oils.—1. Extended experience with various oils has shown the necessity of using a heavy oil to protect the low speed train of double reduction gears against pitting and other deterioration. Heavy oils, however, are not well suited for high speed bearings, particularly where the clearance between bearings and journals is small and will produce high bearing temperature and increase the friction loss. As it is not practicable to use either oils of two different viscosities in the same system, or to install separate systems for the gears and bearings, the only practical compromise is to increase the viscosity as much as possible without heating the pinion and turbine bearing. The viscosity of the oil at the point of entering the bearings can be controlled over a wide range by regulating the amount of water supplied to the oil coolers.

2. In the tropics, it may be necessary to use both coolers to keep the temperature down, and in northern waters, in the winter, it may be necessary to throttle the water supply to the coolers or cut it out entirely.

3. The general rule is to keep the oil as cool as possible without overheating the high-speed pinion bearings, paying also special attention to the bearing and thrust at the forward end of turbines.

4. The temperature may arise to and operate at 160° F. without danger, with a good supply of oil and careful watching.

5. It cannot be too clearly emphasized how necessary it is to keep the oil pure and clear of water, by using the test cocks provided to denote its presence, by consistently heating and settling oil when the presence of water has been detected, and by keeping the oil pressure at coolers at all times above water pressure. Every opportunity should be taken to remove all sediment from the oil both by heating and settling, and strainer baskets should always be kept clean and filter cloths regularly renewed. If this is systematically carried out, the oil should last and remain perfectly lubricant indefinitely and make-up oil be reduced to minimum.

6. By agreement with the Division of Operations, the following revised list of approved oils has been adopted for use on all installations:—

Brand.	*Maker.*	*Viscosity at 100° F.*
D. T. E. extra heavy	Vaccuum Oil Co.	525
Calol extra heavy	Standard Oil Co. of Cal.	700
Ursa	Texas Company	750
Algol	Texas Company	510

7. These brands have been selected with reference to distribution facilities of the manufacturers, as well as suitability of the oil.

8. Should other oils be approved from time to time after satisfactorily passing the required tests and after having proved by actual operations that they are suitable for the purpose intended, they will be added to the above list and all concerned will be duly notified.

9. Under ordinary conditions, the viscosity should be maintained at 450 seconds to 500 seconds Saybolt by keeping the temperature of the oil leaving the coolers at the figures shown below, and which are for 450 seconds and 500 seconds Saybolt viscosity respectively for the different brands.

Brand	Saybolt viscosity	Approximate degrees Fahr.	Saybolt viscosity	Approximate degrees Fahr.
	Seconds		*Seconds*	
D. T. E. extra heavy...	500	102	450	106
Calol extra heavy	500	110	450	113
Ursa	500	111	450	114
Algol	500	101	450	104

DOCUMENT B.

Technical Order No. 75 as Amended 7-16-19.

List of equipment approved for use in connection with standard lubricating oil system.

Filters and separators.	*Type or size.*	*Maker's name.*
Peterson Marine Oil Filter. (See note below.)	100 gallons per hour.	Richardson Phoenix Co., Milwaukee, Wisconsin.
DeLaval special centrifugal oil purifier.	No. 300 motor driven.	DeLaval Separator Co., Poughkeepsie, N. Y.
Sharples centrifugal.	No. 6 motor driven.	Sharples Specialty Co. West Chester, Pa.
B. F. Bowser marine type filter. (See note below.)	100 gallons per hour.	S. F. Bowser & Co., Inc., Fort Wayne, Indiana.

Note.—Filters shall be made of black steel plate at least $\frac{3}{16}$-inch thick to permit of caulking and shall have mill scale removed by pickling, and shall be painted with black enamel after assembly, on outside only.

Oil coolers.	*Type or size.*	*Maker's name.*
Griscom-Russell multi-whirl oil cooler.	No. 116.	Griscom-Russell Co., 90 West St., New York.
Schutte & Koerting oil cooler.	No. 8.	Schutte & Koerting Co., Philadelphia, Pa.
C. F. Braun.	300 square feet.	C. F. Braun & Co., San Francisco, Cal.

NOTE.—Oil coolers to have composition tube plates and brass or copper tubes.

Oil strainers.	*Type or size.*	*Maker's name.*
Schutte & Koerting duplex oil strainer.	4-inch duplex.	Schutte & Koerting Co., Philadelphia, Pa.
Elliott twin oil strainer.	4-inch twin.	Elliott Company, New York.
Coen 4-inch duplex oil strainer.	4-inch duplex.	Coen Company, Inc., San Francisco, Cal.

DOCUMENT D.

Automatic Electric Alarm for Control System—1. *Purpose.*—To prevent the wrecking of turbine and gears due to failure of oil supply, by immediately sounding an alarm and signaling the operator when oil level begins to fall in gravity tank, and simultaneously speeding up the oil pump so as to restore the oil in tank to normal level.

2. *Equipment.*—The system shall consist of two float switches, one installed on each gravity tank; one alarm and operating panel installed at engine operating station, on which are mounted one gong, one red light signal, two white light signals, two automatic throw-over switches for supplying current to the system from generator or radio battery, one double-pole, single-throw, main-supply switch; one double-pole double-throw switch, for connecting separately each gravity tank float switch, one panel installed in chief engineer's room with one gong, one red and two clear light signals, one magnet operated steam valve installed in steam line to oil pumps for controlling oil pump speed; wiring and connections.

3. *Current supply.*—Direct current is normally supplied from the lighting set generator bus, and led to the system through two automatic throw-over switches. These automatic switches have their main throw connected to generator bus and auxiliary throw to radio storage battery. The switches are equipped with magnet or low voltage release, connected across the generator bus, and in the event of a shut-down of the generator automatically connect the system to the radio storage battery, thereby insuring a continuous supply of current.

4. *Connections.*—The alarm gongs and red light signals in the engine-room and chief engineer's room are connected to float switches on gravity tanks, through a double throw switch. This switch is thrown on the particular tank which at the

time is being used as a gravity tank. A clear light signal is also connected across each throw of the switch, in order to indicate that the proper tank switch is in operation and also any interruption in the supply of current to the system.

The magnet operated valve, installed in by-pass line across main throttle valve in steam line to oil pumps, has its magnet terminals connected in parallel with gong and red light signal in such a manner as to open simultaneously with closing of float switch.

5. *Operating cycle.*—When oil level falls about one foot from top of tank, float switch closes, causing gong to ring, and lights red signal lights in engine-room and in chief engineer's room. At the same time magnet valve in by-pass steam line to oil pump is opened, thereby speeding up oil pump, which in turn increases supply of oil until level of oil in gravity tank is again normal.

When oil is restored to normal level, float switch opens, cutting off gongs and red signal lights, closing magnet valve and causing oil pump to resume normal speed.

The pump is normally throttled to half speed, or approximately fifty single strokes per minute. The operating engineer should be carefully instructed to control the speed of the oil pumps by means of the throttle valve installed opposite the branch containing magnet valve, and not from the two valves adjacent to the oil pumps. The latter valve should be used only for cutting on or off the oil pump it is desired to operate or hold in reserve.

Material Specifications for Automatic Electric Alarm System—1. *Float switches.*—Two enclosed float switches, two poles, open circuit operation five amperes (minimum), 125 volts, adjustable for maximum distance of 18 inches.

2. *Tank-head brackets.*—Two tank-head brackets for mounting float switches on gravity tanks. Material, cast-iron flanges to be drilled for 16⅝-inch bolts on 18½-inch diameter pitch circle. Float to be enclosed in tube secured to tank heads and extending down in gravity tank for a distance of 4 feet 9 inches, to prevent improper action of float due to rolling of the ship. Tube to be of sufficient diameter to allow free movement of float, and to contain one bearing support for float rod. All screws, etc., to be securely locked by means of copper wire carried through screw heads.

3. *Magnet operated valve.*—One magnet operated 1½-inch valve for full boiler pressure and wired for 125 volts open circuit operation.

4. *Alarm panel.*—One alarm and operated panel, of black slate, on which are mounted the following:—

One red light bull's-eye signal equipped with one 25-watt, 125-volt, clear Mazda lamp.

Two white lights bull's-eye signal each equipped with one 25-watt, 125-volt, clear Mazda lamp.

One 4-inch gong D. C.

Two automatic throw-over mechanically interlocked switches, single-pole 5 amperes (minimum), 125 volts D. C. solenoid operated and arranged for main and auxiliary circuit supply.

One double-pole, single-throw knife switch, 30 amperes, 125 volts.

One double-pole, double-throw knife switch, 30 amperes, 125 volts.

Brass name plates shall be provided, labeled and mounted on panel as follows:

"LOW OIL" between red signal light and gong.

"GENERATOR" on main or generator connection to automatic throw-over switch.

"BATTERY" on auxiliary or battery connection to automatic throw-over switch.

"SUPPLY" between D. P. S. T. supply switch.

"TANK NO. 1" between one white signal light and corresponding throw-over D. P. D. T. switch.

"TANK NO. 2" between the other white signal light and corresponding throw-over D. P. D. T. switch.

5. *Chief engineer's panel.*—One chief engineer's panel, of black slate, on which are mounted the following: One red light, two white light signals, one 4-inch gong, and to be of the same specification as for main alarm panel.

DISCUSSION.

The President:—Paper No. 14, "Standard Lubricating Oil System for Geared Turbines," is now open for discussion.

Mr. W. W. Smith, *Member:*—Mr. President and gentlemen, this paper contains much valuable information and should be of especial interest at the present time. During the past two years the Federal Shipbuilding Company has built for the Emergency Fleet Corporation thirty vessels. Eighteen of these were equipped with the pressure type of lubricating oil system, and twelve with the standard type of oil system described in this paper. The standard system that we installed, however, could hardly be called standard, because of the very many changes which were made; in fact, I would say that these systems were rather special because no two of the ships were alike.

In general, there are two basic types of lubricating oil systems, the pressure and gravity systems, and both of these systems are considered satisfactory, provided they are designed properly, and properly installed. The following are some of the salient features which are considered essential for satisfactory lubricating oiling systems:—First of all, the system must be simple, and it is considered important to have the control of the pressure automatic, or the control of the system arranged so that it does not require a great deal of attention from the operator. Secondly, there should be provided means which will prevent the complete breakdown of the turbines in case the lubricating system should fail from any cause. Thirdly, there should be an ample margin throughout in the entire design of the system.

I will give a brief outline of the two types or systems which were developed by our department for this work. The pressure system developed is composed of two pumps, controlled by pump governors of the diaphragm type. These pumps discharge into a supply pipe or header which supplies the oil directly to the bearings and gear teeth, having first passed through suitable strainers, etc. Having passed through the bearings and gears, it drains back into a drain tank, from which it is sucked up by the pumps and discharged back into the system. The pressure is maintained by the operation of a pump governor, which causes the pressure to be constant, regardless of any differences of steam pressure or resistances on the oil side, which is considered an essential feature.

In addition to the automatic control of the pressure we provided a safety alarm—a mechanical device—by means of which a small whistle is blown in case the oil pressure should go down below what was considered a safe limit. In addition to this, we also provided a positive safety-stop device for the turbine itself. In case the pressure goes down to about half of the working pressure, this device will shut the steam off from the turbine and actually cause the turbine to stop.

The gravity system which we developed was somewhat different from the system described in this paper, in that the pumps were controlled by a float in the drain tank, so that they automatically keep the oil pumped up into the gravity tank, and in this way they required no attention at all from the engineer. The automatic control of the pump, I think, is quite an important feature, because without it the engineer is likely to fail to regulate the speed of the pump correctly. In spite of some of the safety devices which were adopted, one of the ships which we turned out was broken down, due to the fact that the gravity tank ran dry, the bearings having burned out.

In comparing the two types, the pressure system is considerably simpler and cheaper and requires less piping. The piping is smaller and requires fewer tanks. The pressure is also more positive and is more under the control of the operator. If any condition occurs requiring more pressure, he can adjust his pump governor and get what he wants.

In the case of the gravity system, the maximum pressure is fixed by the designer, but in some cases, through improper pipe design or some fault of design in the bearings, etc., it is desired to get a higher pressure than the system provides for, which cannot be done. This is also true where a very heavy oil is used and the temperature is low. This increases the viscosity and the friction of the oil in the pipes, which cuts down the pressure at the bearings. I saw quite a serious accident occur from this cause.

The pressure system has certain advantages in that it does not require such close designing, and you can carry a pressure of 10 or 15 pounds, which is more than you actually need at your bearings. In other words, you provide a large margin of safety in the case of your pressure, and, in my opinion, that is an advantage for the pressure system which should be considered.

The difficulties which have been experienced with lubricating oil systems have been pointed out in this paper in several places. From my observation, these difficulties were due mainly to faulty installation and partially to faulty operation. In the pressure systems where these troubles occurred, they were nearly always in systems where the pressure was not automatically regulated. If, after the pump speed has been adjusted, the steam pressure should go down, it will cause the oil pressure to go down, and in some cases it has caused the pumps to slow down to such an extent that the oil pressure has failed and the bearings have burned out. Had this installation been provided with the proper automatic regulation, this accident would not have occurred.

I would say that the condemnation of the pressure system, which has been made by a good many people, is based on those systems which were not properly regulated, and which, I think, do not come up to the standard requirements for a pressure system.

There are several points in the paper which I wish to refer to. From its title it might be understood to refer to the single standard type of lubricating oil system. It is believed, however, that other types than the one described in the paper are also good, or perhaps better than the one which is described here.

Another point is, that the object of standardizing, of course, is a good thing in certain cases, but in this case it is believed that the degree of standardization was carried too far and not enough latitude was allowed the designers to produce a good installation. In many cases the provisions of these technical orders were such that they could not be carried out at all. The layout of the ship did not permit it.

In connection with these technical orders, there is one point that might be brought out, namely, a great deal of energy was spent in directing the designers how to build these lubricating oil systems. As far as I have been able to find out, no instructions were provided for the operating force in telling them how to operate the oil system. It seems more important to have the operating force know how to run the oil system than it is to instruct the designers to design them, because in most cases you have a fairly good man to design these systems. That is referred to on page 244 of this paper. We prepared a very thorough set of instructions for the ships we turned out of the pressure type of lubricating oil systems, furnished the instructions to the engineers, and had no trouble whatever with any of them.

On page 245 it states: "Nearly all the oil coolers previously used were inadequate to transfer the required number of British thermal units, which amounted in a number of instances to 10 per cent of the propelling turbine horse-power. This was partly due to the fact that the reduction-gear manufacturers had overrated the efficiency of their reduction gears." That is a matter of design, of course, and it does not apply to these oiling systems where the coolers and other parts of them were properly proportioned and figured out. In this connection, however, attention is called to the fact that if the lubricating oil cooler is designed for a certain viscosity of oil—say 300 seconds at 100° F.—it will not take care of oil of twice the viscosity and give the same result. The heat transfer is affected greatly by the viscosity of the oil. In some ships the systems were designed for medium grade oils, and then very heavy oils were put into them. In such cases the oil systems could not be expected to function properly.

On page 245 it is stated that 10 per cent of the propelling horse-power is transmitted to the oil. I hardly think that is the case. At least I have never seen any tests or experiments that would show that so much heat was transferred into the oil, and I would say, offhand, that about half of that would be the correct figure.

On page 245 it is stated that the technical order specified a drop of 20 degrees in the oil temperature. It is not possible for the designer to fix the temperature drop in the oil cooler, for the reason that the flow of oil is determined by the design of the machinery. There are so many holes of a certain size, and it takes just so much oil to go through it. Then, again, the quantity of heat transferred to the oil is constant under constant speed conditions. Therefore, with constant circulation and constant heat transfer, the temperature drop in the oil regulates itself—the designer has no control over it.

In our particular vessels the temperature drop in the oil was about 12 degrees instead of 20 degrees, due to the fact that there was a very heavy circulation of oil, amounting to about 200 gallons per minute. If the circulation of oil had been 100 gallons per minute instead of 200 gallons, the temperature drop would have been 24 degrees, or just double.

It was mentioned in connection with the size of the cooler that additional area should be provided to take care of certain deposits, etc., in the tubes. I have not found any difficulty from these deposits, and it would seem to me, in case there were deposits in the tubes, the thing to do would be to clean them with kerosene or some other cleansing agent.

The necessity for suction strainers is rather questionable, and they have since been omitted from this order. The size of the mesh in the discharge is given as 1/64th of an inch, to remove the sediment. It is not believed that all sediment can be removed by the use of strainers; certain fine particles will go through and can only be removed by settling and filtering. It seems to me that openings anywhere from 1/32 to 1/64 of an inch are satisfactory.

On page 246 the subject of filters and separators is treated, and under that subject it is noted that the gravity supply tanks are arranged for settling the oil. It is not considered good practice to do this, and it is believed to be better practice to provide a separate tank for settling the oil, so that all the dirt and sludge deposited will, as far as practicable, be kept out of the circulating system.

The alarm systems which were provided for in this order have not given very satisfactory results, and it is believed that it would be better to eliminate the electrical features and provide some purely mechanical method of giving an alarm in case the oil pressure fails.

On page 247 mention is made of the oil pumps becoming vapor bound. This condition in the pumps may be due to several things—of course, hot oil contributes to it—but the condition is aggravated by high velocity and friction in the suction pipe and by high piston speed in the pump. Lubricating oil pumps should be conservatively rated, and our practice is to put in pumps about one-third the catalogue rated capacity.

In regard to the viscosity of the oil, originally the lubricating oil was specified to have a viscosity of 700 seconds at 100 degrees—that is a very heavy oil, which gives rise to certain complications. In the first place, the oil requires very large coolers to extract the heat on account of its viscosity. In the second place, if the engineers allow the oil to get too cool, it becomes very sticky and does not flow properly in the pipes, and in one case I know of a serious accident occurred due to the oil not flowing through the gears and bearings. It would seem that an oil of medium viscosity—one coming between 300 and 500 seconds—is more suitable for geared turbine units.

On page 248 it is mentioned that the first cost of installing this system is quite high. We found it quite high, about double the cost of the pressure system which we put on previous vessels.

There are a number of details and other points in the paper which may be questioned. Most of them are minor details and really matters that the designer should be allowed latitude in handling. In general, this system is considered unnecessarily complicated for cargo steamers, and does not seem to comply closely enough with practical requirements.

Mr. W. L. R. Emmet, *Member of Council:*—I would ask Mr. Fernald to advise what the approximate cost of such a system is.

Mr. Fernald:—It may be put in for about $21,000. We had estimated running to twice that amount.

The President:—Is there any further comment?

Mr. Parker M. Robinson, *Visitor:*—Experience has undoubtedly shown that the oiling system on a geared turbine unit is a most vital factor, and a great deal of credit is due to Messrs. Schmeltzer and Fernald for recognizing this fact at an early date and taking the necessary steps to develop a reliable system.

Technical Order No. 75, which has now become famous, received a large amount of adverse criticism and all sorts of gibes were pointed at it; yet on the whole it was a big step in the right direction.

The question as to whether the gravity or the pressure system should be used was settled in favor of the gravity system, and that seems logical because of its more inherent reliability. The pressure system is used in the Navy, because it is undesirable to have any vital part of the machinery above the protective deck; and in the case of destroyers the headroom necessary for installing the gravity system is not available. However, the pressure system operates quite satisfactorily on these vessels became of the much larger personnel in the engine-room force, and because of the fact that the periods of full power operation are rather infrequent and of short duration.

The service required of a vessel in the merchant marine is entirely different, requiring practically continuous full-power operation while the ship is at sea. This fact, coupled with the small size of the engine-room crew, makes the more reliable gravity system the proper choice.

As to the details of the system described in the paper, there are one or two points on which I would like to comment.

The first is in regard to the strainers. The type of strainer shown is the familiar type of twin-basket strainer. This has many good features, but it has the one disadvantage of not being fool-proof. If the engineer neglects to change over the valves at the proper time, either the flow of oil will be completely shut off, or else the basket will burst and all the accumulation of dirt, together with parts of the broken basket, will be carried on through the system. This will result in more harm than if the dirt had not been separated from the oil at all.

To guard against such an occurrence a by-pass should be installed around the strainer, in which is located a spring loaded relief valve. Then, when the oil pressure reaches a certain point due to the clogging up of the strainer basket, the relief valve would open and the flow of oil continue uninterrupted. To inform the engineer when the oil is flowing through the by-pass, a simple indicator could be provided, and he would then know when it was time to change over to the other strainer basket.

As a further safeguard against misoperation a by-pass with a relief valve can be installed around the coolers, so that it is impossible to shut off the supply of oil by incorrect manipulation of the oil cooler valves.

One other point on which I desire to comment is in connection with the gravity tanks. Two tanks are shown, either of which can be used as the supply tank while the other is used for settling purposes. This requires the duplication of float control devices, heating coils. etc., which could be avoided by having only one gravity tank with a settling tank located just below. Any water or dirt which collects in the bottom of the gravity tank can be drained directly into the settling tank, from whence it can be passed through the purifice and back into the system. This would seem to be a somewhat better arrangement than the one shown in the paper.

MR. C. R. WALLER, *Member:*—The paper presented by Messrs. Schmeltzer and Fernald is, in my opinion, a great contribution to the records of this Society. Three years ago there were very few vessels built in this country where geared turbines were used. Some vessels had been built in Europe using geared turbines as propelling units, thus giving European shipbuilders some experience of the care and maintenance of geared turbines. When the shipbuilders in this country started to build vessels equipped with geared turbines, they had therefore very little experience themselves, and the technical press had not as yet given detailed information covering the European experience. One point must also be remembered. When we started to build geared vessels in this country, we did not plan to build one vessel today and one tomorrow, but we started with a plan to finish one in the morning, about fifty in the afternoon and two hundred the next day. In other words, we started out on a very big scale to produce something that had not been manufactured before.

The trouble first experienced with geared turbines on board these new vessels was found to be due to poor oiling systems, and I therefore think that the work done by the Shipping Board and Mr. Fernald has eliminated a great percentage of this trouble. I do not mean by this that all the troubles that we have had on board geared vessels have been due to oiling systems, but it is a fact that in the beginning the lack of experience with oiling systems for geared vessels was a great drawback.

To give you a little illustration—I visited a shipyard on the Pacific coast in the early part of 1918. A geared turbine was installed in a vessel and an oiling system was also provided for. The engineer who was called upon to design this oiling system very likely did not know much about the problem of handling oil, as otherwise he would not have designed an oiling system in such a poor manner. The oil pumps were placed about 6 or 8 feet from the storage tank in the double bottom, and in the suction line were placed an oil strainer, oil cooler, about one-half dozen valves and a number of 90-degree elbows, so that, if he ever attempted to pump oil with the pumps as they were arranged, the oil pumps would have to be operated with slightly over 24 feet suction lift, which of course is ridiculous. Unfortunately, the designer undoubtedly did not know much about the details of oiling systems, neither did he know that the handling of oil is entirely different from the handling of water.

This paper has brought out a great many points, I think, that shipbuilders, in the future, will take advantage of. I would like to mention a few words about the close-pressure system as compared to the overhead tank system brought out by Mr. Smith. I believe the overhead tank system is far superior, for the simple reason that it gives the personnel in the engine-room a greater safeguard against any sudden failure of the oiling system. Before going on a trial trip on a standard vessel built at one of the shipyards at the Pacific coast, I was trying to convince the shipbuilder to use overhead tanks. The shipbuilder, however, said "No; tanks cost too much money." After considerable persuasion, he installed one tank for handling spare oil. During the official trial trip of this vessel he had six shut-downs, due to the failure of the oiling system. Each shut-down would have been eliminated had overhead tanks been used. As it was, the oil pumps pumped direct, furnishing oil to the oiling systems at about 15 pounds pressure. The pressure was maintained by diaphragm valves, but these valves did not always function properly. With the overhead tank system it is possible to observe if the oil capacity diminishes, and, before the oil supply reaches a dangerous minimum amount, steps can be taken to either shut down the machinery or to start other oil pumps. A great many of the troubles with some of the early vessels were caused by the oil failing, and in most cases it was not discovered that the oiling system was functioning improperly until it was too late and bearings were damaged.

Another feature which I think is important in reference to oiling systems is the facilities for keeping the oil clean and keeping it free from salt water. In connection with the old engine practice, if the engineer discovered he had a hot bearing, the first thing he would do would be to put water on the bearing, and by doing this some water would of course get into the oiling system. It is undoubtedly true that you can lubricate a bearing with 50 per cent oil and 50 per cent water if you have to, but if you try to feed oil and water to a reduction gear, you will very quickly run into trouble. The lubrication of reduction gears is the most important detail for proper maintenance of the gear. It is absolutely impossible to operate a reduction gear where comparatively high pressure is used except by having the proper facilities for lubrication, and the oil must be free from water, as otherwise you are bound to have difficulties.

The effect of water on the reduction gear was plainly demonstrated when comparing the appearance of the gears on board two different destroyers made by two different shipyards. The one destroyer went to sea without any facilities for keeping the oil free from salt water, and after two days' service an inspection of the gears showed that the gear surfaces were coated with rust and the oil was found to contain quite a high percentage of salt water. The destroyer built by the other shipyard went to sea with an oil purifier installed for keeping the oil in good condition. Upon the return of this destroyer, the inspection of the gears showed no presence whatsoever of rust, and the lubricating oil was entirely free from dirt, sediment or salt water.

The same thing is true in connection with merchant ships. I have seen vessels come into New York Harbor equipped with geared turbines, the oiling system containing 60 per cent oil and 40 per cent water, and invariably no one considered the trouble on board the vessel to be caused by the condition of the oiling system. I am sure that the paper here presented will be of great help to the different shipbuilders and will make them realize that, in order to insure satisfactory operation of geared turbines on board vessels, it is essential that an oiling system is provided that will properly and efficiently lubricate the equipment.

Mr. Charles P. Wetherbee, *Vice-President:*—I am only going to say a word. There are two ways to keep a rubber ball round—one is to apply pressure uniformly over it, and it is pretty hard work to do that. The other way is to remove all pressure, which is easier. I think the hard way has been tried here.

This complicated method of supplying lubricating oil to turbines and gears is absolutely unnecessary, and this idea of putting $21,000 worth of equipment into a 3,000-horse-power ship for the lubricating system seems to me to be going pretty strong. It would be a powerful argument against the use of geared turbines if such a lubricating system were necessary. However, it is not necessary, and the very simplest kind of arrangement for oiling of the turbine system with forced lubrication has given the utmost satisfaction, and the simpler things are the greater the satisfaction.

As to the question of getting salt water into the lubricating oil, if you keep the oil pressure in the system higher than the pressure used for the circulating water of the cooler, you may lose oil, but you will never get salt water into the oil.

On one installation with which I am familiar, where the power runs from 25,000 to 38,000, situated in a very confined space, the lubricating oil system, even for that large installation, did not reach $10,000 in cost. The more complications you put into such a system the more your troubles accumulate, and one thing reacts on the other and makes the whole system less desirable.

Mr. Ernest H. B. Anderson, *Member:*—The Shipping Board officials had an exceptionally difficult task to fulfil, and it seems to me they deserve every credit for the manner in which they tackled many problems which required the most careful consideration in connection with fitting out and installing various types of turbine and gear equipments in the freighters building at the new shipyards all over the U. S. A.

None of the shipyard officials had any experience whatsoever with installing machinery in vessels, yet they were called upon to equip and fit out vessels with auxiliary machinery, the successful operation of which was of vital importance to the main propelling machinery. By means of technical orders and diagrams, results were obtained which fully justified such a system.

On Plate 108 is shown the revised method for installing the forced lubrication system to the turbines and gears, and I think it is unfortunate that there is no cross-connection arranged so that the oil can be supplied direct to the bearings without passing through the gravity tanks. In other words, it should be possible to supply oil to the bearings and gears either from the gravity tanks or alternately from a direct-pressure system.

Mr. Smith:—There is one point brought up in the discussion—that the gravity system is inherently more reliable than the pressure system. I cannot quite agree to that point. We turned out eighteen of our ships with pressure systems, and twelve with gravity systems. The only ship we had any trouble with was one fitted with the gravity system—every one of the pressure system ships has been satisfactory. Not long ago I was informed that, on another ship, not made by our company, one of the oilers was sent up to shut a valve. He shut off the valve controlling the supply of oil to the turbines and burned out the bearings. That was with a gravity system. Both of these accidents occurred with gravity systems. The oil pressure can fail on the gravity system as well as it can fail on the pressure system, the only difference being that there may be two or three minutes of time interval between the gravity and pressure systems. In both cases, I think some safety device must be provided to give the engineer warning to shut down the turbine, to prevent its complete breakdown, since both systems may fail.

I have had considerable experience with pressure oil systems during the last ten years, and I think some of the statements were based on hearsay rather than direct observation. It is my observation in every case that the pressure system has worked more satisfactorily than the gravity system on the ships we have turned out. The engineers invariably use the pressure system by the by-passing of the gravity tank in preference to using the gravity system, since the former is simpler and easier to operate.

The President:—It is apparent that where there are many men there are many minds. Does any other gentleman wish to contribute to this discussion? If not, I will call on the authors to make such rejoinder as may be appropriate.

Messrs. Schmeltzer and Fernald:—Mr. Smith's discussion consists of two parts, one a description of the pressure system which his company was using before the Emergency Fleet Corporation adopted the standard system for his ships, as well as all others being built for the account of the Fleet Corporation. The other constitutes criticism of the standard system.

In selecting the gravity system instead of the pressure system we were influenced by the following facts:—

(*a*) The oil companies advised us that the deterioration of oil in pressure systems was largely due to the small amount of oil in the system and to its being constantly circulated and agitated without any opportunity for freeing itself from either air or water. These difficulties could be easily overcome with a gravity system.

(*b*) Lubricating oil of the correct grade and quality was not available in foreign ports, so that it was advisable to have in the system, when the vessel was put in commission, a large quantity of oil as well as provision for keeping that oil in good condition. A gravity system takes care of this requirement to best advantage.

(*c*) The gravity system with large tanks provided a margin of operating safety of from three to five minutes for adjusting any parts of the system which might become inoperative and switching over to spare apparatus without shutting down the propelling turbine. By using the spare gravity tank and the storage tanks and wasting the excess of oil to the bilges, it was possible to operate the main turbines from 20 to 30 minutes with all oil pumps, etc., out of commission.

(*d*) With a pressure system it is necessary to shut down the main propelling turbine, either manually or by automatic devices, immediately on failure of the oil pumps.

There were numerous instances during the war when the propelling machinery of vessels had to operate in spite of all commercial considerations, even to the limit of destroying machinery. A case in point occurred near the Azores. A twin-screw turbine-driven vessel had operating trouble with the turbines due to an accumulation of water in the lubricating oil. A submarine attack was imminent, and the engineer was ordered to operate the turbines regardless of damage to them. This was done, and the submarine was escaped, although the turbines were almost completely destroyed.

The use of a protective device which automatically shuts down the main propelling machinery endangers the vessel itself, because of the unreliability of its operating and permitting the vessel to drift into a dock or on a lee-shore or when trying to escape a submarine, and is generally condemned by operating men. For that reason an automatic stop-valve operated by failure of the oil pressure was eliminated from the standard system. Furthermore, it was considered unsafe to depend absolutely on any automatic devices and, where used, they were supplemented by facility for visual observation and manual operation.

Mr. Smith raises the technical point that the title of the paper is too broad. It is recognized that the system is standard only for vessels of the Emergency Fleet Corporation, and this limitation is clearly brought out in the paper itself.

Mr. Smith also states that no instructions regarding the standard system were provided for the operating force. The technical order promulgating the system contains rather complete operating instructions. Copies of it were furnished not only to shipbuilders concerned but to the Division of Operations, and it was expected that the shipbuilders would provide the usual instructions furnished with vessels and that the Division of Operations would take care of the instructions of the engineers through their superintending engineers and schools. We believe that the expected action was generally taken, although no handbook for operators was ever prepared or published.

The variations in the system causing no two vessels to be alike, mentioned by Mr. Smith, must refer to very small details, because, as brought out in the paper, the only substantial change in the system was made after the armistice, and for the reason set forth in the paper. The use of mechanical alarms and other automatic appliances as an alternate to the electrical system was always permitted.

Mr. Smith refers to several accidents caused by the defects of the standard gravity system. The instance of damage to the bearings on account of the viscosity of the oil is inexcusable because the steam coils in the gravity tanks permitted the viscosity of the oil to be brought to the correct point before starting up the units and was put in both gravity tanks to serve the double purpose of controlling the oil viscosity at starting and aiding in the settling of water out of the oil. The same trouble would have occurred with the pressure system if the engineer had not adjusted his automatic appliances so as to increase the pressure to take care of the extra viscosity of the oil.

The other two instances referred to by Mr. Smith were clearly cases of operating inefficiency. In one case the engineer was sent up to close a grating and, instead, closed off the oil supply from the gravity tank. Ordinarily a sight feed or flow indicator in the main supply pipe from the gravity tank was specified. No satisfactory device was found, and this requirement was later eliminated. It is doubtful if it would have prevented this accident. In the other instances no part of the lubricating oil system failed to function. The vessel was delivered by the shipbuilder with one of his trained engineers to a nearby port for loading. At this point the trained engineer left the ship without instructing the new crew, who took the vessel to sea. For some reason the lubricating oil pump failed, and the turbine continued to run on the gravity tank until it ran dry. At this time the chief engineer was in his stateroom. When he reached the engine-room the alarm was still ringing, and the crew was so green that they not only had not stopped the main turbine but they made no effort to start up the spare lubricating oil pump or to switch over to the reserve gravity tank.

We agree with Mr. Smith that any system may fail and, in the paper, called attention to the limitations of the system in this respect. Even the extra time element for taking care of trouble provided by the gravity system is useless if the crew do not avail themselves of it and ignore warning signals. As Mr. Smith's operating experience with this system is limited to trial trips, and as the engineers are trained at his yard, according to his own admission, by using the pressure by-pass (eliminated in the revision), he cannot expect his criticism of the operation of the system to receive serious consideration.

Mr. Smith's criticism of the 10 per cent loss in the gears is certainly well taken, based on what would be good engineering practice, but, unfortunately, the losses in some of the earlier gears were almost as large by actual test. The excessive losses were due to the use of too close fits in the numerous high-speed bearings.

Mr. Smith criticises the viscosity of the oil selected and recommends one of lower viscosity, but does not submit any reason for this conclusion. The manufacturers of the turbines set the amounts of oil that they required, as well as the viscosities; violation of their requirements vitiated their guarantees. We were buying turbines in large quantities, and (using a word popularized by the Shipping Board) we were allocating them to various shipbuilders, so that we were responsible to them for the turbines, and in order to protect ourselves we had to adhere to the stipulations made by the manufacturers, so that we had viscosities all the way from 280 to 720 to deal with.

As a result of operating experience and the modification of bearing designs, it was possible to standardize the viscosity and pressure of the oil and the size of the pumps, coolers, etc., taking care of the variation in requirements of different makes of turbines by controlling the temperature of the oil in operation and making an initial adjustment of the supply of the oil through reducing flanges or cocks.

Mr. Smith also raised a number of other points which we believe are already answered in the paper or to be merely a difference of opinion, and, therefore, unnecessary to reply to in detail. His condemnation of the system in general, as being unnecessarily complicated for cargo steamers and in complying with practical requirements, comes under this head.

Mr. Robinson's suggestion of an automatic by-pass around the strainers appears to us to amount substantially to eliminating the strainers. The time element required to switch from one basket of the strainer to the other is negligible. If the strainer should begin to clog, the level in the gravity tank being used would naturally fall, and the engineer would have warning of this fact both from the gauges, and, where they were installed, the automatic alarm, and ample time to make the change. The twin strainers all carry a plainly marked plate containing complete operating instructions. His suggestion that the heating coil be eliminated from the gravity tank removes all operating control over the temperature and viscosity of the oil before starting up.

Mr. Wetherbee's comment, based on his experience, which we believe is almost entirely with destroyers, therefore, is naval practice where the service is not continuous and where operation is generally at reduced capacity. His success in building torpedo-boat destroyers naturally gives him confidence in his ability as a designer, and his criticisms naturally merit serious consideration. His sweeping statement that a system of such cost and complication is absolutely unnecessary is, however, directly at variance with the opinion of other eminent marine engineers, some of whom have had as much experience with merchant vessels as Mr. Wetherbee has had with destroyers, and who passed on and approved every detail of the standard system before it was issued. His prescription for keeping salt water out of the oil is a part of the standard system. It is not, however, infallible, and even destroyers have suffered from this trouble, as pointed out by Mr. Waller.

This standard system was initiated by the failure of the simple pressure systems which Mr. Wetherbee commends. We think his comments would be subject to much modification if he were familiar with the character of operating attention which many merchant vessels received during the war, and are still receiving. We had occasion during the war to ask Mr. Wetherbee for some information regarding governors for marine turbines, and he replied most positively that they were absolutely unnecessary. In the case of a turbine designed to operate safely at 25 per cent overspeed and installed in a vessel subject to only slight variation in draught we agreed with him, but a number of our vessels were equipped with turbines which could not operate safely at 10 per cent overspeed and many of them had to return from France without cargo and without ballast except a small amount of water ballast. As the turbines were equipped only with overspeed trip governors, which were not always operative, there was great danger of damage to the turbine under these conditions, and we disagreed with Mr. Wetherbee to the extent of standardizing a type of governor which would prevent racing but would not shut the turbine down and yet permit perfect hand control of the speed. We believe Mr. Wetherbee would have done the same thing under similar conditions.

Mr. Anderson thinks that the pressure by-pass should not have been eliminated from the system. In reply to Mr. Anderson we would say that the pressure by-pass was eliminated as being undesirable except as a war measure. In case of war it could be installed quickly and at small expense.

Mr. Waller's comments require only our thanks for his commendation of our effort.

The authors are glad that the paper elicited so much discussion, even though some of the critics dissent from us in their opinions.

It is unfortunate that more operating men did not take part in the discussion. The system is, as stated in the paper, a compilation of the ideas of many people. So far as we know, nothing has heretofore been printed on the subject, and it was only with the greatest difficulty that any information on many essential points was procured. As a starting point in the perfection of lubricating systems for various services, we believe it will serve a useful purpose, aside from any direct value it may have for the purpose for which it was designed.

In conclusion the authors wish to express their appreciation for the help and cooperation in designing the system and in the preparation of the paper which was rendered by their associates in the Engineering Department of the Emergency Fleet Corporation and the Division of Operations of the Shipping Board, and to the numerous engineers connected with shipbuilding, manufacturing and operating companies for their advice.

Especial thanks are due Mr. A. Conti, chief engineer, and Mr. H. B. Taylor, engineer, of the Emergency Fleet Corporation, for their assistance in the final revision of the system and simplification of the diagrams.

THE PRESIDENT:—I am sure that you wish the thanks of the Society to be given to Messrs. Schmeltzer and Fernald for their very interesting paper and the interesting discussion which followed from the presentation of the paper.

Referring to one of the statements made in the discussion, I do not think we can blame the mechanism when the personnel cannot tell a grating from a valve.

The next and last paper on the program is entitled "Electric Propulsion of Merchant Ships," by Mr. W. L. R. Emmet, Member of Council.

ELECTRIC PROPULSION OF MERCHANT SHIPS.

By W. L. R. Emmet, Esq., Member of Council.

[Read at the twenty-seventh general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 13 and 14, 1919.]

The use of electricity for propelling ships was first advocated in the case of large warships in which it affords particular advantages in the matter of cruising economy through change of speed ratio, interchangeability, space distribution, etc. The first application, however, was made in the case of the U. S. collier Jupiter, which is in most features a ship of the merchant type. The demonstration of geared-turbine propulsion came after the first serious proposals of electric drive, and the advantages which have been attributed to the geared method have suspended such activities as were considered in this country in the direction of electric drive for merchant ships, while, in the case of warships, electric drive activities have been uninterrupted. In the meantime certain electrically driven ships built in Europe, operated with very high degrees of superheat, have shown wonderful fuel economy, and many more such ships are being equipped.

The larger American shipbuilders, having their own facilities for machinery construction, have, not unnaturally, been opponents of electric drive, and the Emergency Fleet Corporation, which for some time has represented ownership, has for various reasons discontinued such activities in this direction as had been planned.

Two or three years ago the writer was of the belief that the geared equipments then being made afforded a solution of the problem which in cost and results would probably prevent commercial success of electric drive in merchant ships, although it was realized that the margin of possible advantage was small. Since that time improvements in electrical designs have been developed, and limitations of gear possibilities have appeared which put the question in a different light, and it is now the writer's belief that electric drive is justified in all large ships and that it will very soon develop a wide application, notwithstanding the great efforts of skill, organization, and capital which have been given to the introduction of the gear drive for vessels of all classes.

The discussion of this subject is largely a matter ot comparison with other methods, and the purpose of this paper is to make clear what is proposed in a specific case and to suggest comparisons which may affect relative value.

The case selected is that of a vessel of 8,800 deadweight tons, length 424 feet, beam 54 feet, having a cubic capacity of 460,000 cubic feet and capable of making 11.5 knots with 2,500 shaft horse-power delivered to a propeller operating at 100 revolutions per minute. Plates 116 and 117 show an electric propelling equipment applied to such a ship, and, for comparison, an equipment with triple-expansion engines is shown in Plate 118. It will be observed that in this design the motor is placed as far aft as convenient, affording space for disassembling and for removal of the tail shaft. The generating unit and controlling equipment are placed near the boilers in such a manner as to afford a maximum convenient saving of cargo space, the condenser being suspended below the turbine in the same compartment with boilers. The auxiliaries are distributed in convenient locations in the turbine room and in the space below near the condenser.

The weight of this equipment, including generating unit, motor, controlling mechanism, and direct current exciter, will be about 67 tons.

Auxiliaries.—In connection with such equipments it is proposed to use, as much as possible, electrically driven auxiliaries. It is necessary to maintain an electrical supply independent of the main generator for purposes of excitation and lighting. The losses involved in the operation of larger auxiliary generating equipment are relatively much less, and there is no increase of complications. With such an equipment it is proposed to install two 150-kilowatt, turbine-driven, direct-current auxiliary generating units, one being required for service and the other installed as a spare. Excitation and lighting will only amount to 40 kilowatts, leaving 110 kilowatts available for any possible auxiliary uses. A little more than half of this should be sufficient for normal conditions. While the ship is at sea, for reasons of simplification and economy, it is proposed to exhaust the auxiliary generating unit into one of the lower stages of the main turbine at a pressure somewhat above the atmosphere, so that some of this exhaust steam will be available for feed heating if that obtained from steam-driven auxiliaries is insufficient. In port these auxiliary units would be exhausted into an auxiliary condensing plant which would be idle while the ship was at sea.

Motor Compartment.—The motor carries the thrust bearing and is also equipped with a simple, slow-moving oil pump which maintains automatic lubrication in the motor compartment. This lubrication can be arranged with a storage tank and with an emergency drip supply to the low-speed bearings contained in the after compartment, so that, even if the oil pump should fail, many hours might elapse before injury could result to any of the bearings. With such an arrangement the self-lubrication of this compartment becomes entirely simple and safe, and with occasional inspection it should be operated without an attendant and without any passage connecting it with the engine-room; in fact, there is nothing that an attendant can do in this compartment, and there would be quite as much reason for keeping an attendant on the truck of an electric locomotive where the electrical and lubricating conditions are far more complicated.

Space Saving.—It will be observed that the omission of the shaft alley and the diminution of space required for the engine-room materially increases the cargo space and simplifies its shape. This increase amounts to something over 12,000 cubic feet, nearly 3 per cent of the total capacity of the ship. The omission of the shaft alley, shafting, and supporting bearings effects a weight saving of about 60 tons, and there will be an additional weight saving in the machinery itself since the electrical equipment will weigh about 9 tons less than the engine equipment for such a ship.

Economy.—If this equipment is operated with 200 pounds steam pressure, 200° F. superheat, and a vacuum of 28.5 inches, the steam consumption per shaft horsepower hour, not including auxiliaries, will amount to 9.5 pounds. Under normal conditions at sea, with most of the auxiliaries driven electrically, this should give a steam consumption for all purposes not greater than 11 pounds per shaft horsepower hour. Such a steam consumption will require at least 30 per cent less fuel for all purposes than would be required by a good reciprocating engine equipment operating without superheat, and even if an equal superheat were used with a reciprocating engine equipment, the gain would still be over 20 per cent.

In this connection it must be considered that large numbers of American ships are now being equipped with reciprocating engines and without superheat, although it has been amply demonstrated abroad that the use of high superheat is practical and economical.

If such a ship were in operation 250 days in a year between California and Australia, burning fuel oil at $1.00 per barrel, the saving in fuel over a similar engine-driven ship operating without superheat would amount to about $17,000, and the increased freight capacity leaving California would amount to 585 tons, which is 7½ per cent of the deadweight tonnage.

Reliability.—A study of the records and uses of such electrical apparatus as is applied in this case will show that the equipment is less liable to interruption of service than any other form of single-screw equipment which is applied to vessels. With such an equipment, however, arrangements could easily be made by which the ship could be navigated about half speed with the main generating unit out of service. This could be done by providing a motor-generator set or rotary converter so arranged that the power of auxiliary generating units could be delivered to the main motor. In an electrically propelled ship, electricity is produced simply for one definite purpose, and the arrangement is simpler and more reliable than shore applications where power is taken from large distributing systems. It is also possible to provide automatic means which, by interrupting excitation, guard against the possibility of serious damage through possible accidents or insulation failures. Such electrical apparatus of the type used in ships is very easily repaired, and, even when damaged, can generally be temporarily connected so as to be operative. The knowledge necessary for such repairs is very easily imparted and is constantly being practiced in our industries by persons who have had little or no electrical training.

Reliability of Gearing in Ships.—To make comparison of such an equipment with a gear-driven ship is much more difficult, since a great variety of arrangements of turbines and gears have been applied to ships of this type. In the matter of reliability, as has been said, the electrical equipment is entirely beyond question, while many evidences of serious trouble and deterioration have developed in geared ships of most types which have been produced. Gears have been very successful in many warships, but these are subject to only occasional short periods of high-power service. In some merchant ships gears have been very successful, and in others most serious trouble has been encountered. Variations of results in similar equipments in different ships illustrate some of the possible uncertainties. Parsons' original gear applications operated with a single reduction, very small diameter pinions, and a large diameter gear on propeller shaft. Some of these have been reported to be very successful, but the gains in economy shown in Parsons' publications are nothing like so great as those accomplished by high-speed turbines with double reduction gears. There have, however, been many cases of failure with gears of this type. In fact there seems to be no type of gearing with which trouble has not been experienced after long service in cargo vessels.

Recent production of so-called "Standard English ships" shows they are being equipped with double-reduction gearing, and at the same time that this change of method is being adopted in England the use of single reduction is being extensively advocated and applied here. Although all the original American equipments in merchant ships were double reduction, the writer has seen a solid-gear, double-reduction equipment of American make in which the gears were badly worn and pitted after 17,000 miles of service, and in this case the proportions of gears are closely equivalent to those which have been adopted in the new standard English ships, and the conditions of design and manufacture quite as good.

These indisputable facts and many others certainly indicate that gearing for ships has not yet reached a state of finished development.

One of the uncertainties of gear operation in ships is illustrated by the very great difference in durability of gears in ship propulsion and in shore uses. In trials on shore, gears have borne without blemish, for equal periods, loads equivalent to approximately four times the average loads which have caused bad destruction of similar gears at sea. This is illustrated by the photograph shown in Plate 119, and the data there given are characteristic of many other similar experiences which have developed.

The reasons for these astonishing differences have never been adequately explained. Plate 120 shows a record taken from a torsion coupling on a cargo vessel operating in ballast in a moderate seaway. This record shows that the torque on propeller shaft varied from zero to approximately 75 per cent overload under certain wave conditions. The effect of bad weather on endurance of gears has often been observed, and it is quite possible that variations much greater than that here shown may at times be experienced. In this case the ship was pitching only 4 degrees. Part of the small, quick variations shown in this record were caused by a transmitting ring which ran slightly out of true in the instrument, but otherwise the conditions were such that the record must be substantially correct.

Another matter of uncertainty in geared-turbine equipments is the matter of temperature in turbines. The operation of turbines in the reverse direction occasions large temperature variations, and temperature variations constitute a fruitful source of danger to turbine structures. Plate 121 shows a record for temperature taken by a pyrometer situated between the nozzle and bucket of the last stage of a marine turbine while the turbine was being operated at normal speed in the reverse direction. It will be observed that the high temperatures shown by that record were produced in an extremely high vacuum by the introduction of small amounts of steam.

A turbine when operated in the reverse direction has a friction loss something like ten times as great as when it operates in a normal direction. In the General Electric shops it has been discovered that reversing wheels of marine turbines turn blue with heat when operated at normal speed in a vacuum of 20 inches. While no definite information can be given concerning the possible effects of high superheat in reversals of a marine turbine, the facts here given indicate that such effects may be serious and should, if possible, be avoided. A turbine which is kept running in its normal direction is not subject to any large temperature variations. The economies incident to the use of superheat on shipboard are very great and cannot long be neglected, although there have been few applications of superheat to American ships. The following extract from a letter from Van Nievelt, Goudriaan & Co., Rotterdam, Holland, illustrates the superheat possibilities in engine-driven ships:—

"We are using during the last five years, in our multitubular boilers, 3½-inch tubes, very high funnels and Diamond blowers, and have no trouble at all in getting sufficient steam. We have practically no leakage at the connections of the pipes and boxes. The original pipes are still in use. The capacity of a 20-ton evaporator is sufficient for supplying feed water."

"Three of our steamers have been running half a year without superheaters with a coal consumption of 24 to 25 tons. After fitting superheaters the consumption was about 22 tons, making a saving of at least 10 per cent."

In electrically driven ships the gain is quite as great as is here shown, and no practical difficulties can result even from degrees of superheat which would be impracticable with reciprocating engines.

Efficiency of Transmission.—The selection for comparison of a ship of low power is unfavorable to the electric drive in the matter of transmission efficiency, the conditions being better for this method in ships of higher power. The generator designed for this case has an efficiency of 95.6 per cent and the motor 95.9 per cent, making the transmission efficiency, including cable loss, etc., 91.6 per cent. In machinery designed for certain high-power ships, the efficiency is as high as 94 per cent.

To determine the efficiency of gear transmission as compared with the figure given above, very careful tests have been made at Schenectady. A 2,400 horsepower ship turbine was connected through two sets of double-reduction gearing to a generator, and the steam consumption was tested at various degrees of load and speed; then the same turbine was connected to the same generator without gearing, and tests were run with the same conditions and the same degree of steam flow. All this was done on a testing stand where conditions are uniform and accurate, the gears ran with perfect smoothness, and all conditions were favorable. Since the comparison gives the loss of two gears, the differences are considerable and the determination should be very close to the correct value. This test showed that the performance of a single gear is as follows:—

Shaft horse-power	*R. P. M.*	*Loss of two sets gearing*	*Efficiency of gears*
2,400	87	250 horse-power	95.0%
1,420	77	160 horse-power	94.7%

In addition to these gear losses, we must also consider the loss in friction of the reversing turbine, which is estimated from reliable data to be 28 horse-power, and we must also consider the bearing losses on about 100 feet of shaft, which in perfect alignment will be 8.5 horse-power. These additional losses reduce the transmission efficiency to 93.5 per cent, leaving only 1.9 per cent advantage to the gearing. With shaft more or less out of line and gears operating under sea conditions, it is probable that the losses given would be greatly increased. Noise is an indication of loss, and most marine gears are at times noisy, while the gears in this test were almost silent. The gears tested in this case were of the General Electric Alquist type, and it might be claimed that other kinds of gears would be more efficient, but it is obvious that, under fixed load and with similar gear speeds and diameters, there could be no advantage in any other type even if it ran with equal smoothness.

Cost.—In the present condition of prices it is very difficult to compare costs, but the cost estimates of the General Electric Company on electric equipments for cargo boats and geared equipments of recent design indicate that the electric is slightly cheaper. If we consider savings in shafting support, shaft alley, oiling system, etc., the saving with electric drive should be as much as 20 per cent of the cost of the driving machinery.

Propeller Speeds.—In ships requiring less than 3,000 horse-power, there is some practical disadvantage in using propeller speeds below 100 revolutions per minute because of the large number of motor poles required if a high-speed turbine is adopted. Studies recently made by the Navy Department and elsewhere have indicated that there is practically no disadvantage in using a propeller speed of 100 revolutions per minute on an 11-knot, 2,400-horse-power ship, but in all cases of electric drive the matter of propeller speed should be carefully studied. In ships of higher power it is not desirable to use extremely high turbine speeds, and therefore there can be no difficulty about propeller speeds. Even in low-power ships, lower turbine speeds could be used if expedient, but this is disadvantageous to a small turbine, and the relative advantages and disadvantages should be duly considered.

Operating Force.—The history of the electrical industry has repeatedly shown that persons who have not used electrical apparatus assume that its operation requires a high order of skill and expert knowledge, and of this assumption we have already heard much in connection with electric drive for ships. A vast amount of experience has repeatedly shown that this assumption is the direct reverse of the truth, and a little thought as to the conditions in electrical apparatus should make the reason obvious. Conductor circuits are much simpler mechanically than pipes and mechanical motions, and electrical machinery is simply a combination of electrict circuits with motion of rotation. The connections are easily shown by diagrams, and little mechanical skill is required to make them. The work of insulation can be so done that, under such conditions as exist in ship installations, troubles which might involve difficulty of repair by unskilled persons are very improbable. In all the extensive uses of electricity in mills, mines, railways, and other industries, it has seldom failed to immediately become popular with the operating forces. In no case has this been more marked than in the ships which have been driven electrically. Large electrical apparatus is generally simpler than small, and the machinery used to propel a ship is in many respects simpler than that which is used to light it. Instead of introducing difficulties in the matter of operating force, the adoption of electric drive will eliminate them and make ships much less dependent upon skill and resourcefulness in their crews.

DISCUSSION.

THE PRESIDENT:—This paper, "Electric Propulsion of Merchant Ships," is now before you for discussion, gentlemen.

E. A. STEVENS, JR., *Member:*—The statement that "in the case of warships the electric drive activities have been uninterrupted" is only true of warships built in this country and of a certain class.

It might look, from what is said in the second paragraph on page 277 of this paper, as if shipbuilders have been opposing the electric drive merely for commercial reasons. This, however, is far from the case, a proof of which is that some of these have advocated the geared turbine, and in many instances these outfits have been built by outside concerns.

Reference is made regarding "the great efforts of skill, organization and capital which have been given to the introduction of the gear drive to vessels of this class," that is, freight ships of about 9,000 tons deadweight. There has been a great amount of talk and work done in favor of the geared turbine, but this has been no greater than that carried on by those who favor electric propulsion.

Regarding the arrangement as shown on Plate 116, the following comments, I believe, would be in order. In the first place it is very questionable whether there is very much space saved. The amount of room taken by the motor in the stern would be very close to that saved in the engine-room; besides, there is no provision for ventilation of the motor room. I have been informed by Mr. W. W. Smith, chief engineer of the Federal Shipbuilding Company, that he had investigated the question of using the electric propulsion for this class of ships, and it was found that this means of propulsion required more room and was heavier than either the geared turbine or the reciprocating engine. The generator being placed above the boilers, as shown, would be very impracticable for four reasons:— First, the space above the boilers would be very cramped; second, the engine-room would be considerably hotter; third, the auxiliaries would be on a different level from the operating platform, which is very undesirable. This has been experienced in many vessels that have had such an arrangement. Fourth, the difficulty of providing proper foundations for the turbine and gears. Very few engineers, if any, would care to go to sea in charge of an electric propelled vessel without some means of communication between the motor and the engine-room. Although a shaft tunnel is unnecessary, it would be necessary, or at least most desirable, to have an alley-way for carrying the electric cables, as these must be protected from damage by freight or water that may get into the cargo space. Furthermore, this alley-way should be large enough for a man to pass through and be able to repair the cables while the vessel is at sea, or partly loaded, without shifting the cargo.

On page 278, third paragraph, it is stated that the motor would be self-lubricating and only occasional inspection would be necessary. This might be the case, but I would consider it very risky to leave this compartment with only occasional inspection, for if any trouble occurred, it would very likely be between inspections and might result in a general disability before it is discovered. It is true that auxiliary machinery, such as the steering engine, is placed at the stern, and only occasional inspection is given; however, the difference in importance between a part of the main propelling machinery and the steering engine is considerable.

I do not believe that any greater economy can be obtained with the electric propulsion than with the geared turbine. If it is advisable to use electric-driven auxiliaries, which I doubt very much, there is no reason why this type should not be used with the geared-turbine outfit and the same economy be obtained.

Regarding the reliability, I can see no advantage in the arrangement suggested in this paper for operating vessels with the main generating unit out of service. Having a motor generator set or rotary converter arranged so that the power of the auxiliary generators could be delivered to the main motor would be quite expensive, especially for a piece of machinery that would be lying idle most of the time. Besides, the auxiliary generators would have to be considerably larger than would be necessary for the operation of the auxiliaries. Trouble at sea with reciprocating engines and their remedies are too well known to be dealt with in this paper. We know that, as far as reliability is concerned, this engine is as good as any. With the geared turbine of the cross-compound type, reliability is assured without carrying any spare machinery. There are cases on record where one of the turbines or its pinion has given trouble and was disconnected. While steaming with one turbine a speed of 9 knots was obtained, while the speed obtained with both turbines in operation was about 11½ knots.

If any water should get into the engine-room and on the electric machinery, the chances of trouble are considerable. Although this has been denied on several occasions by advocates of electric drive, I notice that electrical engineers are very careful to keep water away from generators in stationary plants as well as those on board ships.

I might mention an experience on a trial trip of a certain geared-turbine vessel. This trial took place in a gale, the wind at the time blowing as hard as 80 miles an hour, the seas so high that solid water was taken over the forecastle when the ship was running on a course with the wind about four points abaft of beam. There was also a very hard rain, but it was necessary to keep the engine-room skylight open on account of ventilation. The spray and rain came through the open skylight and landed on top of the gear case. If this had been a piece of electric machinery, generating current at high voltage, the result would probably have been short circuits. It would also have been a source of danger, not only to the machinery, but to the engine-room crew as well. Troubles were experienced with the electric circuits on this trip.

Serious troubles with geared-turbine outfits have been caused by the following reasons: First, improper design; second, poor material; third, poor workmanship; fourth, gears too small for the power transmitted through them; fifth, poor foundation; and sixth, poor operation.

The following statement is made:—"In fact, there seems to be no type of gear with which trouble has not been experienced after long service, in cargo vessels." This statement is only correct in one sense, depending on what is meant by trouble. There is no machinery built for marine or stationary work that has not given some trouble at one time or another. Besides, this statement is not consistent with what is said above in the same paragraph, where it is stated:—"In some merchant ships gears have been very successful, etc." It is also stated that gears for ships have "not yet reached a state of finished development," but what assurance have we that the electric propulsion equipment has reached this finished state.

The question raised regarding damage that would occur to operating turbines in the reverse direction seems to be more theoretical than real. With large direct-drive turbines few accidents have occurred; with a small turbine such as is used in the gear drive these accidents would even be less. This is borne out by practice.

It is stated that the efficiency of generator is 95.6 and the motor 95.9, making a transmission efficiency of 91.6, including cable losses. If this be true, the cable losses must be zero.

Reference is made to loss due to friction of the line shaft bearings which the electric drive would not have as the motor is placed in the stern. The shaft in this case is very short. Experience has been that the stern bearing wears down very rapidly, causing the propeller shaft to come out of line with the line shaft. This is largely taken care of by the flexibility of the shaft, which does not exist when the motor is placed in the stern. In almost all cases of general cargo ships it is considered desirable to place the machinery amidship, in spite of the space lost by the shaft valley.

I understand from Mr. Smith that the electric drive was more expensive than either the gear turbines or reciprocating engine, as well as being heavier and requiring more space.

The slight disadvantage, as mentioned, of using a propeller turning 100 revolutions would only exist under favorable conditions. In rough weather or with a foul bottom these disadvantages would increase.

Regarding the operating forces it is stated:—"Conductor circuits are much simpler mechanically than pipes and mechanical motions, etc." This might be considered so by some people, but others think differently. Steam leaks are much easier to locate and repair than short circuits or electric leaks, and steam-driven auxiliaries are easier to repair than those driven by electricity; furthermore, the former is less likely to be damaged by water than the latter. No more skilled labor is necessary to make repairs, on a ship fitted with geared turbines or reciprocating engines, than is required on one fitted with the electric drive. Conditions existing in mills, mines, railroads and other industries are so different from those on board ship that any comparison would be useless.

Anyone reading the description of the U. S. S. Jupiter and New Mexico would not be of the opinion that the electric machinery used to propel a ship is simpler than that used for lighting, nor would they think it simpler than a geared turbine or reciprocating engine. The U. S. navy colliers fitted with reciprocating engines and geared turbines were operated before the war with merchant crews most of the time, while the Jupiter has always been operated by a navy crew.

Mr. Francis Hodgkinson (Communicated):—Mr. Emmet's paper undertakes to describe an electric drive installation for an 8,800-deadweight ton ship of 2,500 horse-power. In the text of the paper he refers to tooth gearing and compares them to their disadvantage with electric drive.

Mr. Emmet paints a picture of gear troubles and their failures, entirely neglecting to make it clear that he referred to the type of gearing, the manufacture and design of which he had been associated with, and did not refer to those of any other type.

Mr. Emmet referred to cases of gear trouble which he stated was due to the teeth being overloaded. In the whole of my experience I know of no case of Westinghouse gears where there has been a single failure which could be attributed to too high tooth pressures. All the troubles which have been experienced have been understood and may be readily avoided. Experiences during the last year indicate we are nowhere near the limits of the possibilities of teeth gearing. Reference is made to the so-called "Standard English Ship," being equipped with double reduction gearing, and a statement made that single reduction gears are being advocated and applied in this country for a ship of the type referred to in the paper. It is true that single reduction gears have been advocated in this country for installations for which double reduction gears are the more appropriate, which can only be attributed to misapprehension and lack of understanding of the gear problem. No gear troubles whatsoever could be attributed to the fact of the gears being double reduction. On the contrary, the difficulty of accurately cutting a gear of large diameter with its possibilities of distortion render the double reduction gear the more desirable and is far more easily installed. Presumably Mr. Emmet is relating his experience with the particular type of gear his company has exploited, and which we understand has proven quite unsatisfactory.

He refers to the uncertainty of gear operations in ships and their being so much less durable than gears on shore. So far as our experience is concerned, there is no difference, but of course it must be reckoned with that misalignment is to be encountered on board ship, and I would not regard any gear as satisfactory for a marine installation unless it were capable of operating perfectly well with the changing position of a hull. Plainly the type of gear with which Mr. Emmet is experienced is unable to conform to such variations as can the well-known floating frame gear, and local concentrated tooth pressures will occur which may readily be destructive.

Concerning the electrical installation, Mr. Emmet gives figures with some degree of looseness. He refers to the space to be saved by eliminating the shaft alley and says that the space occupied by the shaft alley amounts to 12,000 cubic feet, or 3 per cent of the total capacity of the ship; 5 by 7 feet would be a great sufficiency for the tunnel area, and its length, 143 feet, gives 5,000 cubic feet capacity against Mr. Emmet's 12,000. As the discussion pointed out, the geared turbine installation may be placed aft as in the case of tankers, if desired, which would make reference to the space occupied by the shaft alley as irrelevant.

Mr. Emmet discusses the reversing elements of turbines, but I do not think he knows, and I am sure I do not know of a single instance where a vessel has got in trouble because of the heat generated in the ahead elements while the turbine is running astern. I know of one instance where, on the return from a trial trip, a vessel was stuck on a mud bank and ran full speed astern the whole night with no trouble resulting. Mr. Emmet submits a curve which we understand to show the temperatures resulting from running the turbine full speed in the astern direction, and admitting first 7.6 per cent of full load steam through the ahead elements and then 12.2 per cent of full load steam through the ahead elements, the vacuum in the exhaust being 28.9 inches. The above is rather suppositious, for the text is not entirely clear as to these conditions of operation. If my assumption is correct, we can see no relevance in the matter, for no sane person would purposely admit any quantity of steam through the ahead elements while the turbine is running astern.

The subject of what temperatures are reached in the ahead elements with the turbine running astern is an important one, particularly should the vacuum be low.

Experiments have been made with a 1,500 horse-power, 3,600 revolutions per minute Westinghouse marine equipment, which gave the temperatures shown by the curve on Plate 122.

Mr. Parker M. Robinson, *Visitor*:—There seems little doubt but that electric drive is the next logical step in marine propulsion. Electrical apparatus has been developed to such a fine degree in a great many lines of industrial and power-house work that there should be little difficulty in making the application to marine work absolutely satisfactory. The advantages are, of course, very great for the larger ships, especially the battleships, when flexibility and economy at cruising speeds are considered. However, when applied to a typical merchant vessel of approximately 3,000 shaft horse-power, the advantages are by no means so marked. Perhaps for this service the Diesel electric drive promises the best results.

One of the main disadvantages to the electric drive which Mr. Emmet mentioned, and which has probably been reiterated more often than any other, is the problem of getting the operating engineers to understand the apparatus and handle it intelligently.

The same criticism was made of the geared turbines, and to a certain extent it was true. Marine engineers in general are rather inclined to be conservative and are naturally prejudiced against any new piece of apparatus. The majority of ships driven by geared turbines have been commissioned since the beginning of the war, and in the necessity for haste, vessels have often been sent to sea with engineers in charge of them who had been given no opportunity at all to familiarize themselves with the equipment which they were to operate. It is no wonder that troubles occurred. The only wonder is that there were not more break-downs.

Realizing the situation, the U. S. Shipping Board adopted the wise policy of properly training their engineers. Instruction classes were established at the East Pittsburgh plant of the Westinghouse Electric & Manufacturing Company and at the Schenectady plant of the General Electric Company, where the engineers are not only given a thorough insight into the fundamentals of design, but they are afforded the opportunity of coming into direct contact with every phase of the building and testing of units.

One of the most striking effects of this course of instruction is the change that has taken place in the mental attitude of the men. When they graduate from the course, which extends for about three weeks at each plant, they go back to their ships with a degree, as it were, and in the eyes of their fellow-engineers they are geared turbine experts. Needless to say they do their utmost to live up to their reputation, and they are proud of their ability to keep out of serious trouble.

There is no doubt, judging from the reports which have been received, that the engineers who have gone out in command of machinery, after having passed through the course, have proven themselves to be very much more capable and to have more pride in keeping their engines going than was ever apparent in the untrained men, who knew comparatively little about turbines or gears; and this is only natural to expect. With this example before us, it can readily be anticipated that a similar course of instruction on the electric-drive equipment would bear corresponding results.

In connection with geared-turbine units, Mr. Emmet has brought out one or two points which I think would bear comment. In the first place, it seems rather unfortunate that the comparison should have been made between the electric drive and the reciprocating engine, whereas the main interest today is a comparison between the electric drive and the geared turbine. It is probably correct to say that the reciprocating engine will weigh approximately twice what a geared turbine will weigh of corresponding horse-power, so that on this point the comparison is far from being correct.

Then if we refer to Plate 116, which shows longitudinal sections of the two ships, one with electric drive and the other with the reciprocating engine, we find that the condition as represented is rather fictitious, because instead of having the engine in the center of the ship it could be installed in the stern with just as short a shaft as used with the electric drive. Therefore the space taken up by the shaft alley cannot rightfully be brought into the comparison. It is common practice in tankers to install the machinery in the extreme after portion of the ship.

The next point on which I would like to touch is with reference to the efficiency of reduction gears. The figures given in the paper show an efficiency of 95 per cent in one case and 94.7 in the other. These are undoubtedly correct, but we do not know all of the conditions under which the tests were made. For instance, the efficiency of the gears will depend considerably upon the viscosity of the lubricating oil used. If an oil with a high viscosity is used and the bearing clearances are not increased correspondingly, the friction losses will become quite large, and the indications are that this may have been the condition existing on the tests in question.

As a comparison with Mr. Emmet's tests I would like to quote some figures from a test made on a 1,500 horse-power, single pinion, double-reduction unit to determine the frictional losses. Two duplicate gear sets were used, coupled back to back, with a turbine on one pinion and a brake on the other. The first test was run with oil at considerably over 500 seconds viscosity with a temperature of 97° F., and the losses amounted to 86.5 horse-power per gear, which gave an efficiency of 94.2 per cent. This corresponds very closely with the figures given in the paper. Then, without any changes in the test equipment, a second test was run, this time with an oil temperature of 116° F., which decreased the viscosity to slightly less than 300 seconds. (And, by the way, the gears will run at that temperature without any possibility of scoring.) The results of this test indicated a friction loss of 37.5 horse-power per gear, which corresponds with an efficiency of 97.5 per cent.

The question of the pitting of gear teeth is one which is very much in the minds of marine engineers of late, and numerous attempts have been made to explain its cause. However, it is generally known that the major portion of the pitting occurs on or near the pitch line on the teeth of the driven gear, and an analysis of the situation shows that the metal at this point is under a peculiar combination of stresses. Sliding action occurs in both directions toward the pitch line, alternating with which there is the pure rolling pressure directly on the pitch line. This action is, perhaps in a way, similar to that which takes place in a seamless tube rolling mill, where the billet is being squeezed first in one direction and then in another. As a result, the center has a tendency to break up, so that the piercing plug has comparatively little work to do.

The action on the tooth face is for the sliding action to raise a slight ridge at the pitch line; and then, of course, there is the pressure of the pure rolling action right on that ridge, which gives us the unusual combination of stresses. The result is that small particles of the ridge flake off, thus relieving the pressure. If we could conveniently build a gear with a groove cut at the pitch line for the full length of each tooth, perhaps we would have the solution of the problem. Needness to say, such a procedure is impracticable. Nevertheless experience proves conclusively that a large portion of the pitting action occurs within the first few weeks of service, and these flakes which come off seem to relieve the stress at the pitch line, after which no further pitting takes place. At any rate, the pitting does not seem to induce scoring or detrimentally affect the proper action of the tooth surfaces in any way.

Mr. Emmet has spoken about the heat generated in the ahead turbine, while running astern. Unfortunately, the advance copies of his paper did not reach us in time to permit us to work up the experimental data which we have available. Our recent experiments along this line indicate that this point should not cause any serious alarm.

In the diagram shown in the paper the astern speed is given as 3,600 revolutions per minute, which we can assume is the same as for full speed ahead. As a matter of fact it is impossible to attain this speed in the astern direction, because the astern turbine is much less efficient than the ahead element; and since the total amount of steam available is definitely limited by the boiler capacity, it is rarely possible to attain an astern speed greater than about two-thirds of the full ahead speed.

Experience has proven that, with a reasonably good vacuum, the turbines may be run at their maximum astern speed for an indefinite time without sign of distress. Perhaps the best example of this was the performance of the turbines on the steamship Westward Ho, which are admitted to be of a type more prone to heating than some others. That vessel, as you will remember, was torpedoed, with the result that practically her entire bow was blown away, and the forward bulkhead being thus exposed it was not deemed safe to attempt to drive the vessel in the ahead direction. Therefore she was driven astern for a period of about three days, covering a distance of some 400 miles and arriving safely in a French port. During all of that time there was no sign of any distress or excessive temperatures inside of the turbine.

Mr. John Reid, *Member:*—The time is getting short, and I will be as brief as possible and give you a few remarks on a practical application of electrical propulsion. I suppose I am one of the few men in this room who can speak of having had the electrician get his hooks into him—I put in electrical propulsion, but had to take it out again. That is not to say anything against electric propulsion, but I want to tell you why I failed, because I do not know that it has been brought before you.

Some years ago, about 1908, we looked into the matter of electric propulsion on the Clyde and built a boat between 30 and 40 feet long, and put electrical propulsion into her. The boat was a perfect success, as a sample. Then we cast about for a bigger boat. We had to look. We tried Denny of Dumbarton, but they said they were too busy and could not bother with it. The British Admiralty was friendly and promised to allow us to equip a mine sweeper—mine sweepers were not wanted so badly then as they were later on—so the mine sweepers were cut out of that year's programme and we lost that prospect too. I was connected with the matter in a small capacity and took it on myself to get the boat so we could apply electric propulsion. At that time they were anxious on the Great Lakes to get a Diesel engine boat, and I was not pleased with the idea of putting the Diesel of that date direct on the propeller; so we determined to put electric propulsion in, and that was the cause of all the trouble. The boat was 250 feet long with 750 horse-power, and I think you will admit it provided a reasonable experiment. The electrical work was well done and highly successful in every respect—nothing wrong with it—and justified every prognostication made in its behalf by the people who put it in, but unfortunately we married it with the Diesel engine, and that, we found, was a very bad mesalliance. If you are contemplating putting electricity on a Diesel engine, and putting it in a ship, if you will come to me, I will give you gratis some information that may save you a lot of bother. The trouble is that the Diesel engine will carry no overload, and if you put in a Diesel engine or two and electric propulsion, as we did in this boat of 750 horse-power, and expect to secure this 750 horse-power at the propeller, throwing out all transmission losses, you will get into trouble, because the electric generator and Diesel engine will work against each other; if you have variations of load in the propeller, they will work back against the Diesel engines and they will not come up to their work, and you will have constant trouble with your power plant.

As I say, the electrical equipment was simply perfect and successful, and anyone who watched the thing run would have said, as one of the men who discussed the paper a minute ago said, it was the logical thing to have done. Unfortunately, if you make an experiment of that kind, and fall down on it, for one man who knows what happened you will find ten men discussing it without the slightest idea of what went wrong. We took out the electric features and put in steam engines. That was a failure for electric propulsion, but I am not sorry I made the experiment.

I do think, however, that Mr. Emmet would be wise to consult the marine men a little more than he apparently does in making his designs. Mr. Emmet is a remarkable man, who has had a great deal of difficulty in shoving his head through this fence we have tried to put around him, but he has kept at it and has gotten on famously, but he would have gone faster if he had some of the naval engineers enthusiastic on his side.

In this design here the motor and propelling arrangements are all right, exactly as in the small vessel I refer to. We had the Diesel engine just forward of the bulkhead you see in front of the motor, and the shaft connections were very short, and there was no need of a passageway throwing the whole engine-room into one. But when the machinery is amidships you cannot cut out the tunnel and not put in a passageway. That removes the necessity of the two engine-rooms being connected up, but, incidentally, a horseshoe or other form of thrust block might be put in, but that is a small matter.

With regard to the engine-room, Mr. Emmet has shown an arrangement which one of our members has criticised, but there is no object in criticising an arrangement of that kind—what I mean is, there is nothing against that arrangement, providing you put in the proper supports for the machinery. This is being done in the case of a number of ships being built on the Continent today which are to be constructed with a particular form of electric propulsion. That is not new. It is new from Mr. Emmet's view, and the extent to which he has used it, in his layout.

The generating motors are put in the wings of the ship, and everything in the engine-room is above the boilers, and it is only minor details which prevent this arrangement from being quite correct and suitable. There is no question but that the use of electric propulsion does cut down the space occupied by the machinery—there is no question about it—and you can divide the electric generating equipment into small units, if you like, and I have no doubt the electricians are competent to get up two or three units, to be applied to one propeller, and with a slight increase of expense involved in the separation, but of course that has not the economy of a single turbine arrangement.

In various ways, Mr. Emmet's comparison, while not absolutely correct, is in its scale fairly satisfactory, and my conclusion, based on my own investigations at one time on a large ship, was that, both with regard to space and weight, the electrical equipment had the advantage. But in regard to expense—and, of course, the question of expense is simply a matter for discussion from time to time, because it varies so enormously—but as a matter of expense there was not much to choose between electric propulsion and geared turbines. It was higher, of course, than for triple expansion engines. The best proof of the practical nature of these electric proposals is that a firm like the Cunard Company went seriously into the question and before the war decided to have the Caronia equipped with electric propulsion and take out the triple expansion engines. Then the war came along and stopped it.

You know how cautious the Cunard people are, and there was no reason for their coming to that conclusion if they had not been quite sure that the thing was right. As to the question of water getting down into the engine-room, of course it gets into the engine-room, but you can close in the electric motors with very light equipment. We had our motor in the stern of a small ship, and completely covered, and there was not the slightest prospect that water getting down in the engine-room would cause any trouble at all. These are practical details which can be worked out if you get busy and put electric marine propulsion into service.

Mr. B. G. Fernald, *Member:*—Mr. Emmet's paper is entitled "The Electric Drive," but it is largely given over to a discussion of the failure of the geared turbines. A good many of his statements about the geared turbine have already been refuted, but I think it is fitting to state that the failure of geared turbines, which is in everybody's mind now, consists of the failure of geared turbines on ships either requisitioned or contracted for by the Emergency Fleet Corporation. In the case of geared turbines contracted for by private owners, before the war, there was some trouble, but relatively little compared with the trouble experienced later on the Emergency Fleet vessels.

All this trouble was experienced with a few designs (approximately six). Sufficient trouble had been encountered to indicate conclusively that the trouble was serious and not to be overcome by slight modification in design. Unfortunately manufacture had reached a stage where upwards of two or three hundred units had either been delivered or manufacture—had progressed to such a point as to render radical changes impossible without great delay in completion of the ships for which the turbines were intended.

Under ordinary circumstances no manufacturing company or purchaser would have committed himself to a wholesale reproduction of an untried design, or even complete replacement of the troublesome machinery would not have been very serious or expensive. However, under the conditions which actually existed, the trouble was so multiplied as to become more serious and wellnigh disastrous both from the standpoint of continuous operation of the ships and financially.

Psychologically, development proceeds curiously. Mr. Emmet read a paper several years ago, to which he referred in this paper, regarding the geared turbine, and he was just as optimistic in every way in that paper in regard to the geared turbine as he is in this paper about the electric drive. The previous paper on geared turbines could, in modified form, be read again today and just be just as optimistic in tone. With defects disclosed by operating experience, corrected, the geared turbine will find its field, as the reciprocating engine has had, and will continue to have, its field, and perhaps the Diesel engine and electrical drive will all have their fields, but there are some things about the electric drive which I wish to bring out.

In the first place, the greatest advance in the use of electricity was its adoption as a means of efficient transmission of power from a large economical generating station to a lot of small units located at remote points from the generating station. The problem aboard a ship does not involve the element of reduction of losses in transmission from the prime mover to the point of application of the power. The electric drive will sink or swim on the basis of a balance sheet for the operation of the ship, which will include the space, the weight, the economy of the prime mover, and the economy of the entire equipment, including capital charges, and, what is of greater importance, the reliability of the equipment in operation.

Mr. Emmet makes a great point of reliability and anticipates that people unfamiliar with electricity will deny that it will be reliable and will see all kinds of objections. The high-water mark in reliable operation of electric units, as such, and in systems, will be found in the steel industry, including blast furnaces, where the conditions, if anything, are more severe, and the necessity for continuous operation at least equals that aboard ship, if not greater. Today electric motors are doing everything in the steel mill that steam engines ever did, more reliably, at less cost, and with greater ease of operation. I mean they not only operate continuously, without trouble and repairs, but the force required to operate them is less, and it is easier to get men to do it, and they do not need to know a great deal. All the points that Mr. Emmet has brought out, I know from experience are true, but the electric motors and other appliances used in steel mills are not things drawn up overnight and installed. They represent a long development with many bitter experiences, and the electrical operating and engineering staffs of the steel companies were important factors in the development of design. They have standardized rules. In order to enter the market to sell any electrical machinery to steel mills, you must comply with these rules, which are insisted on by the operators of the steel mills.

I would also point out that many of these reliable electric motors drive rolling mills through a double helical reduction gear of the same general type as that used with geared propelling turbines, and that these gears have given just as reliable and efficient service as the motors.

It has been found, within the last five or six years, that, on account of the dust around the steel mills, it was necessary to wash the air used to cool the turbo-generators, and various protective measures have been worked out from time to time. The development of control apparatus to the point of being foolproof and efficient has largely been carried out by the steel mills—that is to say, they share the responsibility for the present high state of development.

Now in the application of electrical machinery to shipboard use, the U. S. Navy Department has done a great deal in connection with auxiliary apparatus. They have their standardized specifications, and they have met with wonderfully good results, but, as far as the propelling units are concerned, the problem is new. The manufacturers of electrical machinery will have to consult freely with those engineers who know marine requirements. A case in point is this variation in torque of propellers that Mr. Emmet mentions; while I may be mistaken, I am under the impression that almost all marine engineers have been aware of that variation in torque for a long time and understood it perfectly.

Among the points that occur to me which must be considered is that a large amount of air will be required to keep the generators and motors cool. That air will have to be taken from outside of the ship, it will vary in the degrees of humidity, and precautions will have to be taken to see that the intakes are thoroughly protected so as not to take in salt spray.

The mechanical details of the apparatus will have to be given a great deal of attention. As to the electrical details, the manufacturers of electrical machinery have had enough experience, so that I think they will amaze the marine world by the reliability of that end of it, but the bearings must be made so that they can be replaced without employing any trained snakes to crawl around and get at the bolts. That point may seem ridiculous, but an examination of the design proposed to the Shipping Board for installation on twenty-five vessels disclosed a condition which would almost indicate such necessity. In such cases, the designer usually states, long operating experience on land engines indicates it will not be necessary to replace the bearings. This is not quite convincing. When anything happens to a ship you cannot telephone to construction men to come and replace the broken part or throw a switch on to a central power station to run the motors while the accident is being repaired. However, there is no reason why these problems cannot be worked out satisfactorily, provided the marine people and manufacturers of electrical machinery get together and swap experiences and go aboard ship and study the operation of this machinery and the results which are to be produced.

The installation on the U. S. battleship New Mexico is the last electrical installation on a large scale, and some information about the accident which occurred on that ship naturally would be interesting because it occurred on a new ship, to which a great deal of prominence had been given, and while probably easily explained as having started in a trivial cause, it nevertheless was a serious accident from the financial standpoint; in a battleship in action it would have been most serious and shows that some elements of reliability have not been taken care of.

The maneuvering gear, or control apparatus, must be arranged so that, when maneuvering near docks or near the lee shore, the protective devices which would cut off the current from the motor shall not operate. It is far better to destroy machinery than to destroy a ship. There are a lot of small details that should be gone into. However, I see no reason why the electric drive should not have its place just as well as the reciprocating engine or any other type of prime mover, if the question of costs is studied for the particular vessel and trade route and proper attention is given to the details of construction.

Mr. A. P. Allen, *Member:*—I have listened with great interest to Mr. Emmet's paper, especially so as coming from one who has had probably more to do with the development of the electric drive than any one man or group of men in this country, if not in the world. But generally I think that we have not given due consideration to the practical side of our propelling units during the last few years as to its adaptation to the work it has to perform as a marine propelling unit.

While I am not an enthusiast on the subject of electric propulsion of merchant ships at the present time, I do believe that it is quite necessary to encourage the development of this type of equipment to the extent of installing the same on a few of our present cargo vessels.

The successful development of the electric drive will depend to a great extent upon just how much the designers and manufacturers of this equipment are willing to be guided by the experience and be advised by the practical shipbuilders and operators, so that we may not have the many lamentable failures and unpleasant experiences that we are now having with the reduction gear drive, and which might have mostly been avoided if the gear manufacturers had worked in closer cooperation with the ship engineers in this country during the past three years.

To illustrate this situation, I would state that in 1916 I was connected with a shipbuilding organization that built the first American cargo vessels to have reduction gear turbines. the order for these turbines and gears being placed with Mr. Emmet's organization. Many additional duplicate orders followed during 1917 and 1918. After a number of these vessels had been placed in service, experiencing frequent gear troubles, we became fully convinced that while these troubles might be partially due to faulty foundations, poor alignment or faulty lubrication, etc., the main cause was due to insufficient tooth area.

While the satisfactory operation of the turbine and gear was guaranteed by the manufacturers, we naturally felt quite vitally interested in the successful operation of these vessels, and I took particular occasion in 1917, at a conference held with the engineers of the manufacturers supplying these gears, to urge on them that it would be absolutely necessary to increase the gear areas at least 100 per cent; that the fault with the gears was that they had been designed only to care for the turbine load and was merely duplicating stationary plant conditions where the gears were subjected to a constant known load only. My argument at this conference was that the load brought on the gears from the propeller in bad weather would be from 100 to 200 per cent greater than any load the turbine would give and that it would be necessary to redesign their gears accordingly.

My arguments were of no avail, and to make a bad story worse they insisted that the overload nozzles, which permitted about 20 per cent overload, be removed as one of the principal causes of the extreme gear wear, thus reducing the overload capacity to about 5 per cent. Think, gentlemen, of a marine propelling unit with a 5 per cent overload capacity.

Mr. Emmet's later investigations, results of which are shown on Plate 120, corroborates the position that we took at that time, but his investigations came too late to save the shipping interest of the country from losses that are and will run into millions of dollars from delays and replacements.

I trust that my remarks will be taken only as constructive criticism, in that we may not, in the development of this new propelling drive, pass through such unpleasant and costly experimentations that we have had with the reduction gear drive.

It is quite unwise to consider installing the electric drive in more than a few vessels, until sufficient time has elapsed to have thoroughly tested it out in actual operation, to permit it being boiled down to a practical operating unit; and while the advocates of the electric drive may be fully convinced that it will be a perfect running unit, still they assumed the same position at the beginning of the reduction gear drive, with the results of which you all are aware.

The President:—Is there any further discussion on the paper? If not, we will ask Mr. Emmet to make such reply as he desires.

Mr. Emmet:—A good deal has been said on which I will not undertake to comment. Much of the criticism brought against electric propulsion on board ships I have been familiar with for thirty years in every branch of the art to which electricity has been applied. The same things have been said over and over again in connection with other uses where it has been proposed to use electricity. The fact remains that, wherever electrical apparatus goes, it popularizes itself. It involves simple rotation, and rotation is much more simple than mechanical reciprocating movements.

Mr. Smith has been quoted as saying that the electric drive was much more expensive, while other shipbuilders have made comparisons similar to those which were made by Mr. Smith and found that it was cheaper, so that there was a difference of opinion.

The suggestion that the Diesel engine was the solution is another thing that I will not go into; it is rather complicated, but I have been informed by very good authorities that the best Diesel engines were the big, slow ones, that the high-speed ones had generally not proven so reliable, and that if you are going to put a Diesel engine on a ship you had better connect it directly with the propeller. I do not know whether that is true or not, but that opinion is held by people who are very good authorities on Diesel engines.

I think that the Diesel engines might be operated in connection with electricity under certain circumstances of speed and power, and I believe that some such installations have been successfully carried out.

The comparison shown in my paper is of the engine instead of the gear, but, in point of fact, if I had put in the gear, it would have taken up almost exactly the same space. It happens that I had the weights and particulars of an engine, and since there are many types of gears, I thought best to make comparison with the engine, which is still in extensive use, for the propulsion of such ships, and which fact now seems to be quite generally preferred to the gear drive.

No investigation of the oil used in these determinations of gear efficiency was made. We were running with a rather thin oil, such oil as is generally used, and it was heated to normal temperature, that is, the gears had been running a long time, and the oil had taken its normal temperature, so that this test was not peculiar in the matter of the viscosity of the oil.

One gentleman suggested that pitting was well known, and that it was generally on the pitch line, or always so. I suggest he look at Plate 119 of my paper, which is a photograph of the gear, which shows pitting all over the face and very large chunks of metal removed.

The pitting is clearly a case of fatigue of metal from repeated overstrain, and, furthermore, experience with ships has shown that this pitting occurs periodically, and every time you take a pit out of the gear you reduce the effective face of the tooth. Thus the more a gear is pitted the more it is subject to additional pitting.

My statement that gears of all types have shown deterioration in cargo ships has been criticised. That statement is absolutely justified, and my personal observation justifies it; that is, I have seen gears of the very lowest degrees of pressures in which the faces were badly pitted, and subsequently received reports from these ships that the pitting had become very much worse. There was one notable case of a ship in which the lowest pressures were used of any gears produced in this country, and we had a report from one of our engineers to the effect that the ship had been run 40,000 miles, and the original surfaces of the low-speed pinions were pretty nearly obliterated; that is, they were badly pitted all over, and will have to be replaced at an early date. You are concentrating force on the reduced area when the gear is pitted. The point I want to make is this, that this thing is in the nature of strains which come occasionally through the effects of the sea, and the ship runs along well for a long time, and then suddenly there is some condition where the gears get injured. I claim, if a 100 per cent performance of gears is to be expected, that these gears must be so proportioned that they do not pit at all. If they pit at all, under any condition, it means in time they will go down, and if they do go down it is a serious matter. The ship will be laid up to renew them, it will cost a great deal of money, and it would very soon eliminate any advantage of the gear if you had to make many renewals. They should virtually last forever.

The speed in reversing was mentioned. Now the test referred to in this paper was at the full speed of the turbine, but the test I mention, where we got 500° F. instantaneous rise by reversal, was made when we reversed at the normal reversing speed, running the turbine at such speed as is produced in service by the reversing of the turbine. We got 500° rise of temperature at certain points in the main turbine.

Mr. Reid's criticism of this design is a true one. This drawing is not correct and shows a bad arrangement. It was done in a hurry, and there were certain things I objected to, when the drawing was made, but we did not have time to change it. I do not like the arrangement of the motor there. There are four bearings too close together, and they might tend to jam against each other. The motor should be a little further forward and a little more room for flexibility allowed. His criticism, however, of the absence of thrust is not correct, because that little wart on the forward end of the motor is supposed to be a thrust bearing—I think that it is not quite a big enough wart, that it should have been made bigger to make room for the thrust bearing.

The question of putting the turbine above is absolutely beyond criticism. The particular design of turbine shown in that paper could be put up on 2 by 4 scantlings if they would hold the weight. The turbine would run perfectly. We run even our larger turbines on structural iron supports with perfect success, so you can put the turbine where you please, put it in the deck-house or off to one side alongside the boilers, or anywhere where you have room for it. I think it would be rather convenient to place it as we have shown it here—the condenser in athwart ship position, and then the drainage from the condenser is convenient, the delivery of steam from the condenser is convenient, and there is an entire absence of strain to distort the turbine. With the condenser so supported, I am sure the apparatus installed as shown will work very well.

Mr. Fernald spoke of my optimistic views concerning gears in electric drive. It will never do the gears any good to try to explain that trouble does not exist. I think and hope that gears can be made, that we will make them big enough to stand up; but the point is that the record to date has not shown it, and they must be made bigger. If made bigger, it is harder to make them compete with the electric drive. In the first place, the bigger gears are less efficient; they cost more and weigh more.

The matter of air supply has been mentioned and the difficulty of possibly getting salt water into the machinery. One experience we have had on board ship is that we get much cleaner air. Carbon dirt and metal dirt which come out of the streets of a city is what bothers us in generators. In the case of the battle cruiser, which is by many times the largest electrical job ever done in the world—in the case of building the machinery for these battle cruisers, we are not taking the air from outboard. We are going to circulate the air locally and cool it and use it over again, we carry it through coolers somewhat similar to the way in which it is done in the case of automobile radiators, and it may be done in a very surprisingly compact space and with a very slight increase in the running temperature of the motor, and the motor will be cool at all times, except when the ship is in very warm water. This is entirely practicable, and is a tremendous simplifier in the application of this machinery to a warship, for the reason that the difficulty of getting these immense ducts in and out of the machinery space is very great; besides, there is very great risk in the impairment of ventilation by damage to these ducts, or danger to the ship by water going down them in case they are damaged by gunfire or other causes.

As to the matter that Mr. Allen mentioned concerning tooth pressure, he implied that our designs had been criticised before we made them. But I do not think that he meant that. As a matter of fact, no such question ever arose that I know of—they were running ships for a year and a half, and praises were being showered on us in regard to the performance of the gears before we knew we were in trouble or had any idea of it. The trouble came all at once, and very rapidly, when a large number of ships began to navigate the Atlantic at the beginning of the war. Of course, if we could have foreseen that, we would have made the gears a great deal bigger, but we would not have known at any time how big to make them, and I do not know today how big we would have to make them, but maybe, if we made them big enough, it would be all right.

Mr. Elmer A. Sperry, *Member* (Communicated):—As usual, Mr. Emmet, in his paper, "Electric Propulsion of Merchant Ships," has given us much food for thought. It is true that, in the installation and operation of new machinery of almost any kind, unforeseen problems arise. The problems connected with reliability of gears used in ship propulsion are very perplexing. These gears are produced by different makers on both sides of the Atlantic. They are designed with a wide range of tooth velocities and tooth loads. Some have been very successful and others have given serious trouble. The trouble seems to be so indiscriminately distributed among the different makes as to baffle analysis. Gear sets of identical make, and seemingly under identical installation conditions, are divided between the very successful and the very unsuccessful. Some of these gear sets give serious trouble.

It is known in the engineering profession that nothing of this kind happens without a reason, and usually the greater the mystery which surrounds the difficulty the greater and more outstanding is the reason when it is finally run down and located. In solving some of the problems of the gyro-compass I have often sent my staff back on the job with the injunction that the real cause of the difficulty is as big as a barn, and with a little more patience there should be little trouble in locating it.

Further, the case is cited where the loading of the teeth is four times and still working perfectly successfully. Mr. Emmet seems to think that there is something about ship conditions that applies. I am inclined to think he is right, but I do not believe that a gradual application and withdrawal of 75 per cent overload, as is shown in Plate 120, will go far in explanation. The seven seconds marked is simply the natural period of pitch of the ship; but I do think that Plate 120 may give a slight clue to one possible source of the real trouble in the minute periods, which can be very clearly seen on close scrutiny; there being easily traceable a high period sub-vibration of something on the order of five per second, which harmonizes with the number of revolutions per minute multiplied by four, which is in all probability the number of blades upon the screw at the end of this shaft.

Critical periods are likely to exist where little suspected in machinery, especially where this machinery is subjected to rhythmic or periodic oscillation of any kind. The latest German submarine Diesels sent over here have two such criticals. These are very serious. The engines in all submarines also have periods, one case notably occurring exactly at the normal running speed. In studying these periods and their causes and finding means for their entire suppression and elimination, we have learned much concerning this whole phenomenon. Such familiarity as I have with this problem leads me to believe that here is another occasion where criticals are on our track and are the prime cause of all the trouble. However, one might say we have no reciprocating engine connected up in this instance, but a turbine instead. Where, therefore, is the source of oscillation? It is true that in the case of the submarines, where the reciprocating engine speeds are only 350, it is easier to study these criticals than at the higher speeds of the steel-turbine, but in these geared sets in ships there are always two sources of periodical oscillation, either one of which, or a combination of the two, may, as we know, constitute all the cause that is necessary to set up a true critical. Theoretically a critical of large magnitude can be set up and maintained by an almost infinitesimally small source of disturbance when this disturbance occurs in proper period.

Now in every ship's gear installation we have two sources of such periods:—First, the unequal action of each successive blade, as it comes down into stiffer water, or as it passes the stern post eddies when in stiffer water, leading or lagging pitch of a blade, or through any number of possible combinations of disturbances, can easily set up sufficiently periodic oscillation to become the source of criticals; second, the gear teeth impact themselves, but we say these are helical and operate with perfect smoothness. This statement, however, for the present purposes, is nothing less than nonsense, because, as stated, an infinitely small source is all that is necessary; let these gears be cut with the greatest accuracy possible, they still have sufficient inaccuracies to perform the full function of setting up a critical.

When criticals exist, it means that certain mass moments are not only present but are so located and distributed in the whole moment system that a proper inter-relation exists. My conception is that such a distribution doubtless exists and can be found and analyzed completely upon close study of the situation. The enormous over-stresses and general havoc provoked and maintained by criticals correspond almost exactly to the phenomenon that has been observed and that we are here considering, which causes disturbances that have no limit in creating unsuspected stresses and pressures that are repeated with a frequency and vigor that can easily be relied upon to explain break-downs and general mechanical impairments and casualties of almost any magnitude.

Think of an 8-inch crank shaft being dropped in two in an engine, simply because the engine is passing through a critical, perfect steel being exhibited at the fracture. As stated, wherever criticals exist one must look for three things:—First, a source of period; second, revolving moments; and, third, more or less resilience within the system. To suppress criticals all one has to do is to place the criticals and the periods "on an open circuit," as it were; that is, there must be a break in molecular continuity between the source of the period and the mass moments.

A magnetic clutch has been recently devised, which promises to provide most perfectly the "open circuit" and thus constitute a complete solution of the problem. The development of this disconnecting clutch is timely, and it is now being installed and is found to completely suppress the serious criticals that exist in the submarine engines. In this instance it constitutes a disconnecting clutch taking the place of the normal engine fly-wheel. The driven moments, while they are being fully and powerfully driven with zero slip, are yet driven through a medium which is such that all torsional molecular concussion is completely open circuited; in fact, the driven element of the clutch is not brought into any physical contact with the driver. The break in the continuity is complete and thus effectively prevents critical periods from building up or even starting.

It is believed that a judicious disposition of one of these clutches in the system, either between the gear and the thrust block on the one hand and the turbine on the other, or intermediate in a double reduction drive, will insure successful operation of the entire gear system by completely warding off the tremendous over-stresses coming upon the gear teeth and other parts through the subtle though powerful operation of criticals, torsional concussion, or vibration.

THE PRESIDENT:—Gentlemen, I am sure you will permit me to extend the thanks of the Society to Mr. Emmet for his very interesting paper, which has elicited so much interesting discussion.

We have now completed our program of technical papers and discussions. Do not forget that tomorrow morning at 11 o'clock the steamer Chester W. Chapin leaves for the Submarine Boat Corporation works; also that the banquet will be held at the Waldorf-Astoria tomorrow night at seven o'clock. We hope then to have a most happy reunion.

There being no further business before the Society, I now declare the twenty-seventh meeting of the Society of Naval Architects and Marine Engineers as having come to a close. Before formally adjourning, I wish to remind you that the success of this meeting is due in very large part to the earnest and devoted efforts of your secretary and his staff, and I know that you will wish me to extend a very hearty vote of thanks to Mr. Cox and his associates in your behalf. (Applause.)

I now declare the meeting adjourned *sine die*.

TWENTY-SEVENTH ANNUAL BANQUET

OF

THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

HELD AT THE

WALDORF-ASTORIA HOTEL, NEW YORK, N. Y.

SATURDAY EVENING, NOVEMBER 15, 1919

REPORT OF SPEECHES AT THE TWENTY-SEVENTH ANNUAL DINNER OF THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS HELD AT THE WALDORF-ASTORIA, NEW YORK, N. Y., NOVEMBER 15, 1919.

The president of the Society, Rear Admiral Washington L. Capps, after the service of the dinner, called the company to order and said:

A word with which you have become very familiar in the last few years is the term "camouflage." You may have noticed on the program of toasts that at a certain period the president would make some introductory remarks. As originally written for my use, that appeared as "Introductory Address." I changed it to "Remarks," and I am thinking of still further amendments, all with a view to abbreviation.

In making any remarks at all, the president labors under a slight disadvantage, because when one of the members of the Commitee of Entertainment came to his office in Washington a few weeks ago, he remarked very politely and naively that our dinners were fine,—they were even inspiring in a way,—but they were too darned serious. That cut out one line of remarks. This forenoon I had occasion to interview the director of the music. I noted that he had a very choice selection. He assured me, however, that he had positive instructions to liven things up and not let it get too serious. Under the circumstances, the sooner I get through my remarks the better.

One hundred and forty-three years ago there was disagreement between those who live in this part of the world and the country which is so well represented by some of our guests here tonight. A number of very earnest and distinguished representatives of the American people got together and formulated a document. One of the most pregnant sentences in that document is:—"That all men have an inalienable right to life, liberty and the pursuit of happiness."

Now we have been through—all the world has been through—a very serious experience. The armistice was signed more than twelve months ago. Peace is not yet declared. There is an unrest, a difficulty of readjustment which may well take us into another year. We, as professional men, scientific men, have a great duty quite apart from our professional work. As men of trained minds it is our duty to let our influence for the best type of citizenship be made manifest wherever we are, and we should not be so much engrossed in our professional life that we do not take our proper place in the great governmental work of our country; in other words, we should make ourselves acquainted with politics, using that word in its broadest and finest sense as descriptive of the science and conduct of government.

I have no doubt that the period through which we have been passing—through which all the world has been passing—these recent years, will adjust itself, and that all things will come out right in the end. But in order to bring about that happy ending, it is imperative that every man shall show his colors, shall take his own proper part in the upbuilding of his country; and, above all, let everybody show that he has in his veins the richest of red blood of a good American citizen. Therefore, gentlemen, I make no apology for beginning our proceedings with an invocation for the Divine blessing. "In God we trust" has been our national motto, engraved on our coinage from the beginning, and you may recall that, when a few years ago it was temporarily omitted, a storm of protest arose. The second number in our program will be the signing of "My Country, 'Tis of Thee," and we are going to have "The Star-Spangled Banner" before we leave. I need not say more in emphasizing the fact that we take pride, every one of us, in being known as virile, law-enforcing and law-abiding citizens of America. (Prolonged applause.)

The company then sang "My Country, 'Tis of Thee."

The President:—The first toast on our program always is to the one we have chosen to preside over the destinies of our country. At present he is stricken in health. With this toast, therefore, we wish to drink most earnestly to his speedy recovery. Gentlemen, the President of the United States.

The company then rose and drank the toast to the President of the United States, and followed with the singing of "The Star-Spangled Banner."

The President:—Among the other injunctions which the toastmaster received was to keep things moving; in other words, the presiding officer must realize that he is here to carry out instructions. It is not often that a naval officer has the opportunity of presenting an army officer to an audience of this kind, and the president is tempted to make use of the opportunity to get in the usual anecdote at the expense of the sister service. A distinguished officer of the Navy who had seen much service in the Philippines had a story which he always told with great relish. As the army officer who will respond for the army is an old Philippine campaigner, I know he will not take it amiss. In the early "days of the Empire" in the Philippines, the Army was constantly trying to put it over the Navy or the Navy was trying to put it over the Army—always in a fine fraternal spirit, of course. In one of the outlying districts, a very good officer of the quartermaster's branch of the service had a casco or native lighter, on which there was a lot of timber logs; there was also apparently some ballast on the casco. The lighter was leaking badly and the quartermaster feared he was going to lose his timber. So he tied a line to what appeared to be one of the ballast stones and tied the other end of the line to one of the logs, and before abandoning the sinking casco, threw them overboard. The stone floated and the log sank. The Army took this reference to their ignorance of some of the peculiarities of Philippine materials a little seriously at first, but they have long since got over it. (Laughter and applause.)

Here in New York, during the last few years, a tremendous troop transportation task has been accomplished by the Army. Nearly two millions of men have been received in and sent overseas from this port and returned. With the exception of one marine disaster, for which they were in no sense responsible, there were no casualties and no serious hitches of any kind. There was also the closest cooperation in every respect with their brothers of the Navy. With great satisfaction, I recently read a beautiful letter written by the naval officer in command of transports to the army officer in command of the Port of Embarkation, and an equally appreciative letter from the army officer to the naval officer. It was no small task to get these men overseas under the conditions existing and in the satisfactory way in which it was done. Therefore, I think the Society is to be congratulated in having speak for the Army tonight an officer well known to all of you as Commander of the Port of Embarkation at New York, Major General David C. Shanks, U. S. Army.

ADDRESS BY MAJOR GENERAL DAVID C. SHANKS.

Mr. President, and gentlemen of the Society of Naval Architects and Marine Engineers, ladies and gentlemen, I esteem it a great honor that I have been asked to say a few words here this evening, although I could sincerely wish that honor had fallen upon some other officer who could more creditably represent the Army, as far as public speaking is concerned. I am sure that this invitation did not come to me because of any knowledge I have of naval architecture or marine engineering. Nobody could know much less about those things than I do. Unfortunately, too, I know it did not come to me because of any supposition that I could entertain you with thrilling stories of deeds witnessed abroad, because, so far as this war is concerned, all of my service has been at home. The only story I could tell you would be of the prosaic and oftentimes tedious duties connected with the organization and administration of the Port of Embarkation. Sometimes I have felt that the hardest blow that befell any officers or any men in our service were those who were assigned to duty at the Port of Embarkation.

Let me tell you just a little something about one of our camps—Camp Merritt, for instance. It took 6,000 men to run the camp, and they remained there permanently. It was their job to do the drudgery. They guarded the camp, cooked the rations, and handled the supplies day after day, while one cheering organization after another went through the camp for service abroad. Their blood was as red and their patriotism as great as any of the soldiers that went abroad, but they had to stay on the job. (Applause.) So far as they were concerned, they occupied a position analogous to that of the small boy who is left at home to mind the baby while the other boys go swimming. (Laughter.)

A few weeks after the armistice was signed last year I went out to Camp Merritt, and I want to tell you a little incident of what I saw there. Under a fine tree was a new-made grave, and neatly sodded over, a paling fence was around it, and at the head of the grave was a white monument. It was new, I had not seen it before, and I stepped over to see what it was, and these were the words which I found on that monument:—

> "Sacred to memory of our hopes of going
> over which died here, November 11, 1918."

(Applause.)

Now, gentlemen, I will not bore you with statistics of the number of soldiers who went across. The greatest number that we sent in one day was 39,000 (applause) and the greatest number that came back in one month was 347,000.

There were a few little incidents out of the ordinary that perhaps I might tell you about briefly. During the time I was at the port, I had one or two letters rather out of the ordinary and rather touching. One of them was from a mother in Pennsylvania, and it was one of the most remarkable little letters I ever read. Her boy had come to Camp Merritt, en route to foreign service, and then had quit and gone back home. She wrote and said that she had begged her boy to return, but she could not convince him that he ought to do it; she had prayed over the matter, and she felt that she was doing her duty by reporting her boy as a deserter. You cannot lick a people when they have such spirit as that. (Applause.)

There was another little incident, quite out of the ordinary, I think, and quite pleasing in its ending. One day last January, when we tried to rush the return of our troops to the maximum, I opened a letter postmarked Detroit, and the first thing that fell out of the envelope on my desk was a picture, clipped from a newspaper, and the picture represented a ferryboat crowded with soldiers. You never saw so many soldiers in all your life on one ferryboat; and in the front, leaning over the rail, was one soldier whose face was unusually clear in the picture—he was rather emaciated, he looked as if he had been ill, and on the cap of the soldier and down on his blouse, beneath his chin, the writer had made a cross mark, as if to draw particular attention to that soldier. When I looked at the letter that accompanied the picture, it was in a scrawling handwriting, and in the simple words of a schoolgirl, and it read something like this:

"General Shanks: My mother tells me to send you this picture. We think it is a picture of my brother, but the War Department wrote us that he was killed in Flanders last October. But, General Shanks, my mother wishes to tell you she knows that is her boy. Can you help us?"

The picture had been so closely trimmed that the name of the paper and the date of its publication was missing, but the girl had signed her name to the letter. I sent for our transportation officer, who had charge of the list of soldiers returning, and asked him to see if he could not go back and find that name, and within four hours he came back and said that he had found the name. Then we drafted a telegram that, of all the thousands that ever went through the port with my name to them, gave me most pleasure. It told the mother her son was in Greenhut's hospital, that he had been wounded, but was rapidly getting better, and would leave for home within a few days. (Applause.)

Sometimes the letters were a little different from that. There was a soldier at Camp Merritt whose courage failed him about the time he got there. The poor fellow could not write, so he got somebody to write a letter on the typewriter for him, and he put in an application for his discharge. He said he had a family in Oklahoma, that his duty called him back to his wife and children, and in conclusion said:—"To vouch for this, I enclose a letter from my wife, and it will tell you all about it." As he could not read or write, he did not know what he was enclosing, and when I read the letter from his wife, it was something like this:—

"General Shanks: Dan cannot read and you need not tell him what I say, but me and the children need the space he takes and the vittles he eats. If you can make any use of him, please keep him and you are welcome to him." (Applause.)

Now, gentlemen, there were two things that brought about the success of our soldiers abroad. The first was the spirit of the men who went over, and the second was the backing they got at home. The spirit of the men who went over did not come all from the men themselves entirely. Its foundation lay far deeper. It lay all over this country in every home, for you know how enthusiastic was that natural spirit, and you know that, whether you were in the highways or the byways, or in the country or the city, that spirit prevailed everywhere.

I wonder if you have ever thought what made that spirit—why was it we had such a fine national spirit. There was one thing more than all others combined that gave it to us —the draft law. As long as our enlistments were voluntary, as long as we had a haphazard man from this point, or that point, or another point, the interest lagged, but when the draft law came along and put every man eligible for military service into that service, the interest of every family became acute, and that is the foundation for our national spirit, and how fine it was! Let us not forget that it was because every man was called on to do his duty, and should the time ever come again when we need men again, let us not forget that it was the draft law this time, and it will be the draft law the next time, that will put upon every man, under every circumstance, the requirement of doing his duty. (Applause.)

Just one word more about the port and its organization, and I am through. It was a sort of threefold arrangement we had. The Navy ran the transports, officered the ships, convoyed them, and looked to their safety. The Army brought the troops into the port, put them into the camps, saw that they were supplied with all the supplies and clothing that were necessary, and put them aboard the ships. There was another organization of which we hear far too little, headed by that great and fine administrator, Mr. Franklin, with a civilian committee, whose duty it was to load the ships and look after the swift turn around. Now these three departments that were interlocked in a system of coordination had to work smoothly, or things would have gone to smash immediately. To me, in all the years that I shall live hereafter, the thing that I shall remember with most pleasure is not the number of troops that went over, it was not the amount of business that was done, but the thought which will forever give me the most pleasure is the fact that these three departments at this port worked smoothly and harmoniously throughout, and there was never a time when we could not adjust things ourselves; there never was a time when it was necessary to go to Washington to smooth out any wrinkles. Admiral Gleaves and Mr. Franklin were such fine men, you had only to say to them it was thus and so and they would say, "All right, that is satisfactory to us." That, to me, will always be the pleasant recollection—I mean the harmony and coordination and fine spirit that existed throughout between these three departments of the work at the Port of New York. I thank you. (Applause.)

The President:—Obviously the Army and its representatives are fully equal to the occasion. Our next toast is "The Navy." Until last evening the name printed under this toast was correctly placed. Unfortunately, I received notice from Washington, late yesterday, that the Secretary of the Navy was compelled at the last moment to deny himself the pleasure of being with you. He sent a letter this afternoon, however, by one of our colleagues who has just come from Washington. The letter from the Secretary of the Navy is as follows:—

"WASHINGTON, 15 *November,* 1919.

"DEAR MR. PRESIDENT: It was with the greatest regret that I found myself unable to be with you tonight. Since becoming Secretary of the Navy I have regarded it as part of my official duty to accept every invitation to the annual dinner of the Society of Naval Architects and Marine Engineers, but, unfortunately, this time there is a conflict of duties which prevents my coming to New York.

"It gave me the greatest pleasure last year, immediately after the armistice, to congratulate the Society upon the wonderful work accomplished by its members during the war. During the last year there has been no let-down on your part, and the progress has been remarkable.

"I am one of those with an abiding faith in the future of the American merchant marine, and how could I fail to have such faith when I know what has been accomplished constructively by the naval architects and marine engineers of the United States, practically all included in your membership? If the problem of the merchant marine were one of construction alone, it would have been already solved. There are future problems, but none harder than those which are largely behind us, and I feel sure there are none that will not be solved by American genius, enterprise and determination.

"Sincerely yours,
"JOSEPHUS DANIELS."

The enforced absence of the Secretary is greatly regretted, but his expressions of commendation of the work of the membership of the Society are deeply appreciated. Two words with which we have all become familiar within the last few years are the simple ones "Carry On." In all well-regulated military and naval services there is no such condition as "no commanding officer" or no one to respond to the requirements of such a position. The one who will respond in this case became our guest with the implied promise that he would not be called upon to respond to a toast. Yet when I learned yesterday that the Secretary of the Navy could not be with us I telephoned him the conditions and suggested that, as the senior naval officer present, he should respond to the toast, "The Navy." He immediately replied:—"I will be at your service." It is a very great pleasure and privilege to have with us the one who will respond to this toast. He really has more years to his credit than your presiding officer, though he may not look it. To tell tales out of school, he entered the Naval Academy several years before I did, and as he was a first class man when I was a poor forlorn "plebe," he was naturally regarded as somewhat of a superman.

During the war he had the great good fortune to command our naval forces in French waters, and recently he was selected by the President of the United States to command our great Atlantic fleet. His splendid record in these two positions of great responsibility is known to all of you. Gentlemen, it is with great pleasure that I give you the toast "The Navy" and present to you Admiral Henry Braid Wilson, United States Navy, Commander-in-Chief of the Atlantic fleet.

ADDRESS BY ADMIRAL HENRY B. WILSON.

The greatest asset of the Navy is, in my opinion, its large number of true and loyal friends who know its work, its aims, and to whom it does not have to be continually justifying its existence. Of these friends, a goodly number are before me, and, therefore, I will not speak on the Navy in general but on that part of which I am the representative—the part which operates the ships you design and construct; and let me say right here that that portion of the Navy believes you build the best ships possible.

Not only are we content that, with the passing years, have come developments of hull design which have resulted in obtaining more speed with ships of greater mass without proportionately increased power; developments in engineering features that a generation ago were not dreamed of; development of the gyro compass and its adaptation to other control instruments; improvements in steering gear and gun gear; but that at the same time you are improving the really vital features of the ship as a fighting machine, you are always ready to give careful consideration to the ideas and wishes of those who live on board and make the ship their home.

It is, naturally, to the latest products of he naval architect and marine engineer that we point with pride and, yet, splendid fighting machines that they are, I sometimes think that it is from the older and, perhaps, the humbler ships that we obtain the best examples of that honesty and staunchness of construction which have stood us in such good stead. Unfortunately, the late war gave us little opportunity to test our large fighting ships, but it did our smaller vessels, and of the lot I know of none which made a name for herself like that of the little destroyer Stewart, built about twenty years ago and of only 450 tons displacement. Prior to the war she had been considered as having outlived her usefulness and destined to be scrapped, but with the submarine campaign in full swing and threatening a victory for our enemy, every craft which could go to sea and steam faster than a submarine and carry depth charges was pressed into service.

The Stewart, with others of her class, eventually found her way to the Atlantic coast of France and was assigned to the duty of escorting convoys up and down the coast—a hard, exacting and most important duty.

During the dark night of March 16, 1918, while the up-coast and down-coast convoys were entering Quiberon Bay, a 10,000-ton British steamer was run down by another vessel. The crash of collision was heard on the Stewart and she was headed at once for the scene. On arrival, it was found that the crew had deserted the ship and that she was rapidly sinking. The crew was forced to return on board their vessel. The Stewart was secured alongside, and both vessels were then headed in the direction of the beach, the Stewart furnishing the power. Here she stayed until the steamer, sinking, parted her lines; but, fortunately, she had been gotten into comparatively shallow water. This large vessel was salvaged, repaired and placed in service again and did some good work before the ending of the war.

One month later the Florence H.—a steamer bound south in convoy, carrying 2,200 tons of explosives for the French Government—caught fire. Soon she was enveloped in a tremendous sheet of fire. The Stewart headed over to the burning ship, accompanied by other vessels of the escort. A few minutes afterwards, there was a series of explosions on the Florence H., and great masses of flaming cases and wreckage were thrown out on the water. The cases began to explode, shooting flame and gas into the air. The entire surface of the sea for a considerable space about the ship was not only afire but in eruption. In the midst of it the Florence H. was aflame from stem to stern. Cries for help could be heard. The Stewart, going at 17 knots, was the first on the scene. She pushed her way through the burning mass to the rescue of the crew, in spite of the fact that on her stern were stored many depth charges. For his splendid work the captain, Lieutenant Haislip, was given the Croix de Guerre by the French, and two men of the crew who jumped overboard and rescued some helpless men were given the Medal of Honor by our Government. The conduct of the officers and men of the escorting ships—especially the Stewart, which took the lead—on that night made a new, separate and glorious tradition of its own for future members of the service to emulate.

On April 23, 1918, the Stewart got her submarine. This submarine was getting in position to attack the southbound convoy which was being escorted by the Stewart and other vessels—American and French. It was a fine piece of work.

Then, after the war was over the Stewart returned to this country under her own power and, I venture to say, in better condition than when she left for overseas.

And so, when we go to sea on the splendid fighting ships of today which the talent of you gentlemen has made possible, we have no doubts about the new and untried mechanism, as to whether it will stand up under the stress of battle. We remember with pride how the older ships which once were new and filled with improvements proved themselves, and feel confident that these ships of today and of the future will bear themselves equally well.

The President:—The destroyer whose performances Admiral Wilson has told you about so graphically was built in the good city of New York—that ought to bring a response. (There was only faint response.) We are nearly all from out of town, I see.

Gentlemen, in August, 1914, there happened a very important event which practically saved civilization. Had the British fleet not been ready immediately to do its duty, Heaven alone knows what would have happened. Not only was it ready then, but it increased in efficiency as weeks and months and years went on. To the unflinching devotion to duty, at all times, in all weathers and under all circumstances, of the men who commanded and served on those ships, civilization owes an inexpressible debt of gratitude. (Applause.) The heir to the British throne is now visiting this country, as did his grandfather about sixty years ago. We have with us at this table distinguished representatives of that Navy which acquitted itself so magnificently in the world's great crisis. Gentlemen, I ask you to rise and drink the health of Great Britain's King, George V. (Applause.)

The company then rose, drank the toast to King George, and joined in singing "God Save the King."

The President:—As in all such world-wide crises, the glory is not confined to one nation. There were many allies, and all participated in the glorious achievements of the war. But, as Britain furnished the first and greatest *naval* bulwark, so France, loved by all, may be regarded as representative of all the allies in *military* operations, since she bore the great brunt of the fighting on land. It is therefore our privilege to rise and salute all the Allies and drink a toast "To France, wishing her all possible prosperity." (Applause.)

The company rose and drank the toast, and joined in singing the Marseillaise.

The President:—It had been the hope of our Society that this toast would have been responded to in person by the representative of France in this country, who also is Dean of the Diplomatic Corps. He is a man of rare attainments and great personal charm, and I deeply regret that you will not have the pleasure of having him here. In a very graceful note, however, he expressed his great regret, and I take the liberty of reading it to you.

"Washington, *October* 21, 1919.

"My dear Admiral: I apologize for my delay in answering your most kind and sincerely appreciated invitation. I was in hopes to see my way clear and be able to absent myself from Washington at the appointed date. Greatly to my regret, I find that my obligations here will not allow me this pleasure and honor, and I am, I assure you, very sorry not to be able to meet those naval architects who have done wonders and put the American Navy in such shape that it could transfer to France enormous armies without the German submarine being able to touch any of the transports conveying troops to France, but only to reach a very few semi-empty ones on their way home.

"Nothing would have pleased me more than to be able to express to your fellow-members of the Society of Naval Architects the gratitude of my country and the sincerest congratulations for the splendid work achieved by them.

"Believe me, with best regards and sincere thanks,

" Very truly yours,

"(Signed) Jusserand."

In the autumn of 1917, affairs were going very badly for our associates in Europe. There was at that time in Washington a gentleman who held a very conspicuous place in the government service and was also one of my most esteemed colleagues on duty with the

Emergency Fleet Corporation. He was "found missing" one morning, to use a Hibernianism. He had left for parts unknown. A week or so later he turned up in the European battle zone, as a very important member of a special Interallied Commission. That gentleman is with us and will respond to the toast of "The Merchant Marine." I have very great pleasure in presenting to you a gentleman very well known in New York, but, as most of us appear to be from "out of town," I present to you the Honorable Bainbridge Colby. (Applause.)

ADDRESS BY HON. BAINBRIDGE COLBY.

Mr. President, ladies and gentlemen, I have no intention tonight of regaling you with the mode of treatment of the merchant marine, with which you doubtless, during the days of the war, became more or less familiar. I shal make no announcement of the number of ships actually delivered into service last week, nor will I indulge in any prophecy as to the tonnage to be delivered in the weeks or months to come. That phase of the merchant marine, if nothing else did, received, from my colleagues on the Shipping Board, the most ample treatment during the piping days of actual hostilities.

I feel that I cannot even approach the very few remarks that I shall make on this subject without referring to the very distinguished chapter of public service which is to the credit of your president. When the history of America's effort to bridge the Atlantic with cargo carriers is written, a strong emphasis will be put upon the work of Admiral Capps in the early formative days of that great effort. (Applause.) I have the more pleasure in referring to Admiral Capps' great part in that work because he never refers to it, he is quite content that no one should refer to it, and I regret to say that there has been altogether too little reference to it. (Applause.) The admiral came upon the scene just as the famous controversy was ending. The redhot sirocco had encountered the simoon of the desert, and honors were easy, when there appeared a quiet naval officer, a scientist in the true sense of the word, a prodigious worker, a quiet and indefatigable and confident presence moving upon the scene of personal rivalry and discord, and the effect was magic. Sir, it was you who plotted the work; it was you who took the initial steps; it was you, with confidence in your grasp and power, who made the great commitments which the Government afterwards carried out; and when you had spent all the strength that you had, when you had given of yourself without stint and without measure, you were forced to lay aside the work as beyond what physical strength your ardor had left to you. But your work was done, and the impress of your power and genius was left as a source of inspiration for other men to draw upon. (Applause.)

It is a pleasing thing that the great events of the war have so far receded into history that a most attractive, if consequent and following, phase of the war has now come to the front. I refer to these personal and human incidents that will come to notice in such increasing volume as the years go on. We will tap that great reservoir of fine humanity to which every nation has contributed, without depleting the sources, until our lives are over.

We have heard tonight, with deep relish, this beautiful story of the heroism of the Stewart, and the fine account of the simple, modest heroism of the stay-at-home regiments that General Shanks has given us. I rejoice that our lives are to be enriched by these anecdotes and tales of heroism and tales of modest worth—these tales of unrecorded bravery.

You referred, Mr. President, to my trip abroad. I wonder if I shall be neglecting too much the subject that has been assigned to me if I pause for a moment to tell you a nautical anecdote of that ship. We were crossing in the old cruiser St. Louis, at about 8 knots an hour to conserve fuel, and we had a rendezvous in longitude 17 West and latitude 48.30 North with some destroyers, sent from Queenstown by that splendid figure in the war, Admiral Sims. (Applause.)

I was very much entertained on the way over by the professional talk of the wardroom. I was led to believe that the submarine danger was really pretty well under foot, that it had been well mastered by the science of the American Navy. I was assured there were three elements of safety: The first was the zigzag course. Then, of course, there was the darkened ship, which was an element of safety, but after all was said and done, it was really the speed factor that spelled safety. They said:—"We will conserve our fuel until we get to that rendezvous," and this longitude 17 West and latitude 48 North began to take on a concrete quality in my mind. I felt I should never forget that point in the ocean, if I once got there —that I should have the topography of that portion of the ocean's surface indelibly graven on my recollection.

We got to our tryst and there were the destroyers, on time to the dot. We started to crowd the fuel on, and the old St. Louis hit it up to almost 17 knots. I was rather disappointed, because I had been reading through some literature which I found in the captain's cabin, which said that she was rated at 22½ knots per hour. A man in the wardroom said: "That does not mean anything except that some time in the long distant past she made that speed without anything in her for four hours on a trial trip, but nobody ever expects her to do anything like that again." (Laughter.) I said:—"How about 17 knots in this submarine peril?" He answered:—"Well, it would be better if it was 19, but I think we will get by on 17."

I felt the ship vibrating and could see from the way the smoke was carried away from the funnels that we were doing better than 8 knots. I felt the rush of air across my face, and everything was going splendidly until we got a few leagues south of the Scilly Islands. You all realize that was the point where a greater tonnage was sunk than anywhere else—it was a veritable ship charnel house.

Something suddenly happened on the old ship St. Louis, and she came to a dead stop. She just eased herself gently that way (the speaker indicating a slightly rocking motion with his hands) between the swells of the sea. I said, "This is awful." I thought of those 8 knots an hour as not to be despised after all, and that it was really preferable to a stationary situation south of the Scilly Islands.

The good ship Huntington, on which Admiral Benson was traveling on the same mission that sent me abroad, signalled our ship. I was on the bridge at the time and said to the executive officer, "What does the Huntington say?" He replied, "She asks 'What the hell is the matter?' " In the meantime our little fleet of destroyer escorts performed antics like a lot of inebriated porpoises. It was a wonderful sight. I asked, "Why don't we send an answering signal?" The officer replied, "We are looking for it, but cannot find it." I then asked, "What are you going to say?" He replied, "We are trying to find if we have any flags that will say 'Damned if I know.'"

This state of affairs, distinctly not on our program, lasted twenty minutes, when the engineer came up from below, covered with grime, with some object in his hand which I did not at once recognize, but it seems that the starboard water feeder had blown its head off and the engineer was carrying the head. We made some improvised repairs and got under way, and the story had a happy ending. But I am told that the starboard water feeder on the St. Louis has blown its head out about every three months since the ship has been in commission, and it was the unanimous opinion of the crew, and I may say also of the three passengers on the St. Louis, that the delay was altogether the fault of the Navy in not carrying an extra supply of starboard water feeders—at least on that ship.

It was a very interesting trip. I remember one very interesting moment to which I have never made any reference before, and I hope that the admiral will protect me in any impropriety that I may be committing by referring to it now. We met the British Cabinet on a day never to be forgotten in the Prime Minister's House on Downing Street. On one side of a long table, with Mr. Lloyd George in the center, was seated every one of those great men who have covered the English name with such luster during the war—Curzon, Milner, Reading and Northcliffe, Gen. Sir William Robertson, the Early of Derby, Bonar Law and all the others. I remember the gracious way in which Mr. Lloyd George opened the meeting. He alluded to an event which occurred some 140 years ago, Mr. President. He said, looking at us Americans:—"This meeting would be very interesting, my friends, were it only by reason of the place in which it occurs." I wish I could even suggest to you the deliberate and impressive way in which this accomplished speaker pronounced his words. He continued:—"It was in this very room, indeed about the very table where we are seated, that some hundred and forty years ago Lord North, then the Prime Minister of His Majesty, King George III, stirred up a great deal of trouble for your ancestors." (Applause.) Then he quickly added:—"But I think I should observe that he stirred up a great deal more for himself." (Applause.)

I remember on our return it was my duty to attend an executive session of one of the Senate committees to lay before the Senate information that had been imparted to us in the strictest confidence by the British Admiralty, as to the true extent of the submarine destruction, and the figures were, indeed, appalling. We had no realization in this country of the extent of the destruction of commercial tonnage by the submarine, and it was not deemed advisable that the true extent of the submarine's ravages should be disclosed to the world. But it was considered not only permissible but proper that the Senate should have the truth before it for its own guidance. I remember making as simple and unvarnished a relation of these facts to the Senate Committee as I could—the facts were so glaring and plain that they needed no embellishment. It was not necessary for any man to endeavor to present them in other than a plain and straightforward way, and I could almost see the pressure grow on the temples of the men who constituted the Senate Committee, who were shocked at the figures, until finally one of the mid-western senators, whose name I shall not, on this occasion, venture to mention, said:—"Mr. Colby, I don't believe that." I may remark that this has some bearing on what I shall say in just a word about the merchant marine. He said:—"Mr. Colby, I don't believe that." I replied:—"Senator, that is no concern of mine. I am responsible only for the truthful relation of these facts to you, and not responsible for your mental reaction upon them." (Applause.)

So, my friends, in dealing with this complex and very scientific problem of our merchant marine, one of the things that makes the problem complicated, one of the things that hangs with heavy menace over the prospect of finding its solution, is the fact that scientific thinking, looking to accurate, well-devised measures, must filter through dogmatic and ignorant political media until we arrive at the problem with our thought and efforts maimed, reduced, and almost neutralized by the necessary passage through that media. (Applause.)

The problem is essentially a scientific one, and yet we approach the problem in the grip of old obsessions, and old prejudices, and old party catch-words. The Republicans and Democrats alike in Congress have a horror of the mention of ship subsidy, and yet we all know in some form or other every nation that has preserved its place on the sea has had recourse, direct or indirect, frank or furtive, to a definite subvention of its merchant fleet. There is under discussion today, and there is an attempt to allay any opposition to the measure by the very indirection with which it is proposed, a scheme of through bills of lading and interweaving ocean freight rates with railroad charges, which is nothing more or less than an indirect and covered, a masquerading ship subsidy.

Another difficulty is that the subject is approached with the bias of some special interest. I do not use "special interest" in the sense in which it is used by the yellow journals or the reform press—I mean an honest and a natural bias. The shipping man is more or less desirous of obtaining government tonnage at bargain-counter prices if he can. The ship operator is naturally interested in an unfettered freedom in the manipulation of ocean freight rates, thinking intently on operating profit. The shipper, on the other hand, is only interested in the reduction of freight rates, and is indifferent to the effect of low rates upon the amount of available tonnage for public service.

This problem is not susceptible of solution by any rough and ready theorem. It is a problem which must be confided to experts whom the public is willing to trust and support, and the solution will be found in the day-to-day contact with and study of the problems which are constantly shifting and are made of innumerable factors, essentially economic and financial. I thoroughly agree with the remark of the Secretary of the Navy that the problem of the merchant marine is no longer a problem of construction; it is a problem of economics, it is a problem of non-political and scientific study, a treatment of difficulties, of situations, of factors, that call for the exercise of educated and unhampered brains. (Applause.)

The solution will grow like a coral reef; it will grow from day to day as the day's problems are solved with enlightenment, and solved without the harassment of preconceived notions or prejudice. We must not expect to arrive at a solution of the problem of our merchant marine by thinking it cannot be touched except in terms of complete respect for the great doctrine of protection, or that it cannot be touched without complete respect for the paramount rights of the mid-western shipper; it is a problem, in the solution of which we should go to nations who are proficient, with open minds and without prejudice, and learning the lessons of their experience, apply them with courage and scientific purpose to our own particular problem, and in that way I believe we can solve the problem. (Applause.)

The President:—Had the earlier part of Mr. Colby's otherwise most felicitous address been submitted to the presiding officer in his capacity as censor, it might have been abbreviated. I will not deny, however, that it is very pleasant to have nice things said about one occasionally. I therefore beg to be permitted to express my appreciation of the gracious terms in which Mr. Colby referred to a certain period of my public service, no matter how undeserved those references may be, and beg also to thank my colleagues and associates for the kindly way in which they received them.

We are going to effect a little transposition in the toast list. The members of this Society are so popular that we are never sure we have their engagements quite nailed down. In this case, one of our principal speakers is compelled by official duties to take an early train out of town. It is needless for me to tell you anything about him. If I told you all I know about him, you would not go home until long after morning. Suffice it to say he is a charter member of this Society, has stood with us in good days and evil days, and is loved by all. I have the great pleasure of presenting to you Rear Admiral A. P. Niblack, who will respond to the toast "The Merchant Fleet in the War." (Applause.)

ADDRESS BY REAR ADMIRAL A. P. NIBLACK.

Mr. President and gentlemen, this introduction has taken me quite by surprise, because I thought that the president was referring to some really elderly person beloved by everybody. I have been a member of the Society since its foundation, as has Admiral Capps, who served as its first secretary, and I have worked my way up through the hawse pipe to the table where the captains sit, but in the good old days there was an added privilege attached to it—we were given free liquor. (Laughter.) In the good old days the people who were seated on the floor of the banquet hall had speeches handed to them and paid for their own drinks. (Laughter.) I was a little surprised tonight, when they put on the small bottles, to find that they contained something different from what we had been accustomed to.

I was reelected one of the vice-presidents of this Society at the meeting of the Council the other day, and when I got a summons to come on to speak on this particular subject of "The Merchant Fleet in the War," I came with the greatest willingness, as I feel I owe it to this Society to tell it some things in the nature of secrets, which should no longer be secrets, about the merchant marine in the war. You will pardon me if I go into a few general statistics, because I feel it is important to tell some things in connection with the work of the merchant marine in which this Society is deeply interested, and which no one has yet told.

I have the honor of representing the Navy at this moment during the visit of the Prince of Wales to this country, and I had the pleasure yesterday of listening to his words, before a group of 2,000 midshipmen at the Naval Academy, of generous praise of the American Navy. I delivered to him an invitation from the President of the Society to be present here tonight. He regretted he could not come. I told him I was billed to speak, and he asked me the subject on which I was to speak. I told him it was on "The Merchant Fleet in the War," and he said:—"It is a very important subject, and I am glad you are going to speak on it." I saw him last night on his way to get a rest after his busy visit to Washington; but I want to say that I got the impression that he is really resting up for the strenuous time he is going to have in New York, and not so much because of the one he had in Washington.

In opening my remarks I want to pay my tribute, in response to what His Royal Highness had to say of the American Navy, to the British merchant marine, and not for the moment of the British Navy, so ably represented here tonight, because no words of mine can paint that picture as it should be and has already been painted.

The British merchant marine in the four years of the war lost 9,031,886 tons of shipping and 14,000 seamen. Of the number that participated in the war, 1,519 of them are on the Honor Roll for bravery in action or in the face of the enemy. (Applause.) The losses of the British merchant marine were terrible. There is a reason why our own merchant marine did not suffer so cruelly, so in what I am going to say I want rather to speak more of the American merchant marine than of the allied merchant marine in general.

It happened that, when we entered the war, the British Navy had the German fleet by the throat, so they were powerless to interfere with the development of our reserve force which we should have had in normal times ready to draw upon. The fleet needed every man and every officer in it, and it would have been very poor policy, on the eve of our action, to take officers and men out of the ships to man auxiliary ships, but we were able, through the hold of the British Navy, to increase our Navy from the first of April, 1917, when there were 5,000 officers and 63,776 men, to 31,186 officers and 501,300 men at the time of the armistice. This increase in the Navy represents really the increase in the number of men assigned to the merchant marine, because we manned, as you know, those ships with naval crews and naval officers.

I had the good fortune to serve in the fleet during the first six months of the war, and we had four squadrons for the training of these men for the reserve fleet. The particular squadron I commanded trained the men for the engineers' force, and we developed a new type of personnel in the form of engine driver, the chauffeur rather than the machinist. The three other squadrons were devoted to training the men for gun crews, radio personnel, and deck force.

At the time we entered the war against Germany, the British and our allied friends had grappled with the problem of the submarine and were just coming to the convoy system as the solution to the problem of getting tonnage across, but prior to that the allied problem had reached the point where they had put one gun on each merchant ship and two trained men to the gun, *i. e.,* they had the pointer and the trainer. The rest of the gun's crew were made up from the merchant crew. We started out on a more ambitious programme of having twenty-one men on each ship for the gun crews and having two guns.

The sinkings in July and August of 1917 became so terrible that it looked like disaster, and the Allies suddenly resorted to the convoy system. Now in thinking of the convoy system we only think of the Atlantic. All of us have been fed up on this. Since returning from abroad, where I spent a year and a half in the Mediterranean, I have been made Director of Naval Intelligence, the particular business of which is to get all information of use for the next war, but I have also been given charge of the historical section to write up the Great War. I will therefore, from this source, summarize a few facts which are worth your attention.

In the Atlantic ship convoy system, there were 1,474 convoys consisting of 18,633 ships, of which 70 per cent were British bottoms, 27 per cent American bottoms, and 3 per cent French bottoms. Information usually stops at this point, but it is interesting to follow the matter a little further. Out of the total number of ships that crossed, only 15 per cent were troop-ships and the rest were cargo ships.

Pretty generally, in the newspapers, reference is made only to the passenger-carrying phase of this work. I am here to speak of the freight part of it, that part represented by 85 per cent of the tonnage, which is not commonly mentioned in that total of convoys.

I was sent over in October, 1917, to relieve my friend here, Admiral Wilson, in the Mediterranean, when he went to the French coast. At the time I arrived in the Mediterranean this convoy system was being put into operation, which required that every ship, instead of going on its own hook, should be formed into a convoy at Gibraltar, which was the main port for the Mediterranean, and which thus suddenly became the greatest convoy port in the world. Of the method of forming the convoys we all know, but the strain that it put on the captains of the merchant ships is not realized. We had to take the captains ashore before each sailing and have a regular school. They had to have rehearsals. They had to learn to handle codes. They had instruction of all kinds, but they had to learn for themselves to keep their position in formation at night at sea without lights, and that is a very difficult thing to do even for ships of the Navy. The merchant marine captains thus had thrown on them a terrible lot of new problems and responsibilities.

It is all well enough to draw word pictures of the modern merchant marine captain and his splendid seamanship, but when four officers of the Navy boarded his ship on arrival at Gibraltar and one inspected his radio, another lined up the crew and gave them a health inspection, then the gunnery officer inspected the guns and ordnance material of the ship, and other inspections were made, the captains began to feel something in the nature of rigid supervision. Then he had to attend the classes in the schoolroom, so that he might be fully advised as to what was required of him in convoy in the exigencies of war. After all this the convoy started out into the danger zone. There were many sinkings, many more cases than we have heard about, for I think probably there were more sinkings in the Mediterranean than in the Atlantic, because of the presence of the submarines, based in the Adriatic.

One of the things I want to say is something that has not yet appeared in the public print in any way, and that is regarding the convoyed cargo ships which passed between England and the Mediterranean. There were 273 convoys between the Mediterranean and the United Kingdom, of which 200 were escorted entirely by American ships, based at Gibraltar, consisting of 4,269 ships, representing 12,000,000 tons of cargo. There were 48 convoys that went from the United Kingdom to Port Said and the Far East, which were escorted entirely by the British Navy and which did not touch at Gibraltar, and there were also 25 convoys which came through escorted entirely by the British Navy from the United Kingdom to Gibraltar, compared with the 200 convoys escorted by us.

In addition to that work, there was, in the Mediterranean convoy system, a total of 10,466 ships, representing 30,000,000 tons, escorted by American ships based at Gibraltar, either singly or in conjunction with the Allies. Of these ships, many were taking supplies to the expeditions in Mesopotamia, Palestine, and Salonika, to Italy and to North Africa, where the French were fighting the natives in the pay of Germans, and also carrying supplies to our troops in France, which went in by way of Marseilles. Much neutral trade was also included in the convoys.

Before closing this rapid sketch, I will mention several things about the submarine problem. When the war came, Germany had only 28 submarines and during the course of the war she built 340 others. The Germans lost 200 submarines during the war, and at the end of the war they had only 168 left, so that the submarine menace was a real one. There is a distinction made always on the other side between what was called the "danger zone" and simply the area in which submarines were operated. The "danger zone" meant where the submarines were actually operating, and you had to have double or treble escorts to get through that danger zone, whereas when you passed outside that zone the escort was reduced to fewer ships.

There is one thing also I want to emphasize. I have just told you the number of ships engaged in the Atlantic troop transport service, and I said that 85 per cent of the shipping was cargo ships. None of those statistics include the ships in the Mediterranean, because there was only one convoy which came from the United States to Gibraltar during the whole war. The supply ships which came to the Mediterranean, as far as North America and South America were concerned, came across singly and unescorted to Gibraltar, and were there joined up with the convoys going on into the Mediterranean.

In the brief history of the convoy system, there is a statement which I think is well worth quoting, showing the difference between the troop and cargo convoys. It is as follows:—

"In general, transports were assigned a destroyer escort, which was about three times as strong as the escort assigned to cargo vessels. In some cases of particularly valuable transports the escort was ten times as strong as the escorts assigned to cargo vessels.

"During the darkness the periscope of the submarine is useless, and submarines must come to the surface if they desire to deliver attack. While on the surface submarines become subject to attack by destroyers surrounding the convoy. During the ordinary dark night visibility does not exceed half a mile to a mile, and as the convoy is completely darkened the probability of a submarine finding a convoy at night is extremely remote, and there is considerable risk to the submarine if it decides to attack. For the foregoing reasons troop convoys in general, while in the open sea, were brought through the most dangerous submarine areas during darkness.

"As only about 15 per cent of the vessels in Atlantic convoys carried troops, it became desirable, so far as practicable, to route troop transports in special lanes through which cargo convoys did not pass. This greatly increased the safety of troop transports, as it practically forced submarines to concentrate their efforts in the areas through which cargo vessels (comprising 85 per cent of the shipping) passed.

"If a German submarine took station in a troop transport lane he might have remained for weeks without sighting a troop convoy. This failure to find shipping in troop lanes forced the submarine into the cargo lanes and so gave a large measure of protection to troop convoys. If the submarines had known the position of these troop lanes and had concentrated on them, they would have found a relatively small number of ships, all of high speed, hence difficult to attack. Furthermore, they would have encountered a destroyer escort three times as strong as the escort protecting cargo vessels.

"Having taken the foregoing steps for the protection of troops, it was practically certain that submarines would be forced to confine their attacks almost exclusively to cargo vessels, and this was borne out by the experience of the war."

In other words, we have all heard of the wonderful troop transport service. All the passenger ships were used for the transport of troops, and the shipments of supplies were made on the slower vessels. Any ship which could make 15 knots went into the troop transport business, and the slow ships bore the brunt of the war. The captains and crews of those ships thus took their lives in their hands and, in convoys at night, were in constant danger of being run into. They were anxious times. Suddenly, in the middle of the night, a ship would be torpedoed, and in some cases the ships sank inside of two minutes, with all hands on board. These men had to bear the strain of the war in a most terrible way. It appears to me incredible that such men as the average merchant ship captains were able, in so short a time, to adapt themselves to these new conditions and to meet the situation in such an excellent way. They deserve all credit for it. I therefore want to take my hat off to the merchant captains of the war. (Applause.)

It will be some years before these facts come out. Whatever you see in the press is always in praise of the passenger service. It was the merchant captains who really bore the burden of the war in this remarkable way. The supplies had to go through, they were in slow ships, and they stood the brunt of it.

There was a fund started in England as a tribute to the British merchant marine, and donations to this fund were asked in this country as a tribute to the British merchant marine, much to the regret and much to the displeasure of the British people. They felt it was not a proper thing to be raising money in America for a fund for the British merchant marine. The plan for this fund has now been enlarged and turned into a fund for the British and American merchant marine, to take care of the widows and orphans and the people who were left destitute from the loss in the war of their husbands and sons on whom they were dependent, in these terrible sinkings, and I, for one, feel that if the American merchant marine has been fortunate in escaping the heavy losses that the British sustained, it is because we learned to counteract the submarine menace by means of the depth charges and the convoy system. I thank you. (Applause.)

The President:—It is a great privilege and honor for this Society to express its appreciation of the magnificent work performed by the merchant marine in the Great War. I am sure that we are all very glad that Admiral Niblack could be with us and, on our behalf, pay this tribute. Coming from a naval officer, with his personal knowledge of conditions, it is especially appropriate.

Our next toast, gentlemen, is "Shipbuilding and Commerce." This Society is rich in presidents. In this case we are calling upon the president of one of our greatest shipbuilding companies. He is also president of the Chamber of Commerce of the United States. He is too well known to all of you for me to say more. I have the greatest pleasure in giving you the toast, "Shipbuilding and Commerce," and coupling with it the name of our most esteemed colleague, Mr. Homer L. Ferguson.

ADDRESS BY MR. HOMER L. FERGUSON.

Mr. President, ladies and gentlemen, it would not be very difficult for me to tell you how much I appreciate being back with you again at an annual banquet after an absence of two or three years, and yet on this particular occasion, my duty, while a pleasant one, I know you realize is a somewhat difficult one. We shipbuilders have, for so many years, had so many grouches of one kind and another until very recently, that you will appreciate that in the presence of the Navy and of the Shipping Board and of the new shipbuilders, it is very difficult for an old-time, has-been shipbuilder to speak his piece.

We are all of us, this year, my friends, able to afford a good dinner, and we hope we will be able to come again next year and perhaps the year after. After that God only knows what will happen. The ships are built, the ships are operating, and we are told of the thousands of tons started out and how many will be launched this week, and next week, and the week after next, but we are not told a great deal about the possibilities of future building.

We who build ships know they are built for men to operate, and we know that they must be operated profitably, or they will not be operated at all, certainly not for a considerable length of time, except perhaps as in the case of the railroads, at government expense; we know even that, after a while, may pall on the American taxpayer. The problems of shipbuilding and ship operating and of commerce are interdependent, with the shipbuilding end of the problem absolutely dependent on the amount of freight, goods and people carried and the commerce done by the owner. We simply satisfy him as best we can—sometimes it is a fearful job.

We have heard a great deal during the war of the new shipyards and of the standard shipyards, and all that sort of thing, and I personally want to pay my tribute as an engineer and shipbuilder to the tremendous work that was done by the Emergency Fleet Corporation and by those people who built these enormous yards in such a very short time, and are now turning out ships at a very rapid rate. I think, for instance, Hog Island is one of the greatest performances that anyone could imagine. I do think, of course, in our shipbuilding, that we are somewhat like General Shanks with his regiments at home—the war ended a little too quickly for us really to get into it. But it ended happily, and I want to assure our British friends that we went into it with both feet and both hands, and that we were going to become prepared to lick these gentlemen all by ourselves, and without assistance, if it could not be done in any other way. (Applause.)

Any man who does not think we went into the war on that scale wants to take a look at the tremendous improvements which took place not only in shipyards but all up and down this coast, in the great ports, where the Government spent millions, without number almost, in preparing for a war of indefinite length.

The problem of commerce is tied up with the problem of shipping just at this time in a rather unfortunate way for us. For instance, American shipbuilders have been abroad to sell ships, and they might have sold ships—and I dare say they did sell a few—had it not been that the rates of exchange makes a ship cost the foreign purchaser about 25 per cent extra, for which he gets nothing. That is a problem of trade which affects our business at this time very deeply.

Many years ago I became convinced, personally, that the shipbuilding problem and the shipowning problem were largely political, that a great many of us were working to build ships and were going to Washington at times with a feeble plea that absolutely had no effect. Up until the time that the Emergency Fleet Corporation was formed, when the Government was forced to go to enormous expense to build a fleet, there was not written into a law, enacted at Washington, a single phrase or a word or a line asked by an American shipbuilder or American shipowner within the last twenty years. Then all at once we became frightened, and we were frightened too, and we sat up nights, and the people sat up with us. Why? Because we found that a marine was absolutely necessary to us, and we also found that the ability to build a marine was absolutely necessary, and I emphasize that point, gentlemen, because frequently people say:—"Let us buy our ships where we can get them the cheapest." During the war we could not buy them, and did not get them the cheapest; we had to have them, and before we are finished we will have spent nearly four billion dollars to get a marine which might have been purchased for one-third of that price previous to the war; and so we broke in seamen, the eamen's law fell down, and the Navy manned the ships, and of course we made mistakes—every one makes mistakes, in fact, they say that is why they put rubbers on lead pencils. (Laughter.)

But shall we make that kind of a mistake again?

We sent to Europe 2,000,000 splendid soldiers, and out of the 2,000,000 25 per cent went in American-built ships, 25 per cent went in German-built ships, and 50 per cent were carried over by our Allies. Is that exactly the position that the people of the United States desire to be put in? Is that what we ever want a recurrence of again?

A man who is a shipbuilder or a shipowner is at some disadvantage, as Mr. Colby said, in advocating his case, but I would like to say that some of us not only like shipbuilding as a business, but expect to keep on building ships. The other day a gentleman said to me:—"What will you do when the grass grows in the yard as it did some twenty-five years ago?" I replied:—"We will cut the grass" (laughter and applause) because we believe that not only are we in a business which, in itself, from an engineering standpoint is the finest business there is, but we believe we are in a business which is absolutely vital to the safety and to the independence of our country. (Applause.)

As your president has told you, I have gotten into work in the Chamber of Commerce of the United States. In doing the work connected with my office in the Chamber of Commerce in the United States, I have been thrown into contact with thousands of business men and have tried in a feeble way to carry the message of a marine and shipbuilding to people back in the middle west and the far west. I also want to tell you that you will find, when you talk to business men, that they want to understand, and they want to know, and in most cases they do understand and know. There is only one sure way of appealing to those who represent you and me in Congress, and that is to appeal to the sound Americanism and sound business sense of the people who send our representatives to Congress, and by no other means will you ever enlist the whole-hearted sympathy and support of the people of the United States in their own marine. (Applause.)

We have the shipyards, we have the skilled men, we have the designers, we have the credit, we have the money—all we lack is the purpose. It was said during the war that never again would these things be allowed to lapse. The trouble with the shipbuilding question and shipowning merchant marine question is that it is too large to be considered from the particular standpoint of any particular group of men who may represent us in Washington.

But the difficulties of securing legislation are certainly no greater than they were in the passage of the Regional Reserve Act, which was probably one of the finest pieces of legislation we ever had, and which stands now between us and trouble, and which, after many years of effort, was enacted.

Last year our total exports from the United States were over seven billion dollars, and there was a balance of trade in our favor of four billion dollars, a tremendous increase. There is now manufactured in the United States from 30 to 35 per cent more stuff than we can use, and it is vital that markets be found for that material or this tremendous production will recoil upon us, which will produce unrest, based on hunger, which is not the case at present, and which would be a whirlwind compared with the gentle breeze of today. It is therefore necessary, not only that we keep up this production, but that we find markets for our manufactured products.

There is a lot of difference between selling raw materials which our friends must come for, and selling something that is manufactured. Copper, cotton, and such things men must come for, because they must have them, but when you come to manufactured goods, they must be marketed, and who but knows that the finest salesman today in the world is a ship bearing the flag of the country from which the ship comes, and in this case it is to be a ship built, equipped and manned by Americans. (Applause.)

The destruction wrought by the German submarine emphasized shipbuilding to a point which is hardly realized. When nine million tons of ships are sunk in a single war, belonging to a single country, how much more important does the ability, both in men and material, to produce ships amount to than it ever has before in the history of the world?

A nation now to maintain itself is absolutely bound to keep at hand those things which make for shipbuilding. A fleet of merchant ships can be wiped out almost over night, and the ability to reproduce those ships is vital to a country. We hear a great deal about "no more wars." There may not be any more—soon. We hear a great deal about this great

League to maintain the peace of the world. I would like to say, gentlemen, that if the peace of the world is to be maintained, it is necessary that the great partners in this enterprise of maintaining peace be partners in reality, and it is necessary, for people to be partners in reality or nations to be partners in reality, that each of them should be just as free and just as independent in the exercise of their rights in that partnership as it is possible for them to be; and with all kindness to our friends across the sea, it must be patent to them, as it is patent to us, who think about these things, that this great country of the United States of America must never again be left so that she must get behind the shield of the battleships of Great Britain or any other country—and she must never again be in such a position that she cannot transport her soldiers where she would send them, and any conditions which bind us so that we cannot exercise our free right of sovereignty and independence is an intolerable condition and will not make for peace, but will make for disruption and discontent on our part.

Sometimes you tell an American audience:—"Yes, the George Washington is a fine ship. That is named George Washington after the father of our country, but it is a German ship from stem to stern." I want to say, gentlemen, not only as a shipbuilder, but as an American citizen, that in the case of the next big war we get into, we at least should have a conveyance of our own making for our own Chief Magistrate. (Applause.)

When you tell the people the real reason, as near as you can, of why we must maintain this thing that has been fostered and fashioned, of why we must keep on the seas, and why it is necessary to ward off any attempt to hamper our own commerce, you will find that the people are willing to go to any proper length in order to secure that result. For instance, take the railroads of the country, now a most burning question. What brought them under the control of the government? Twenty-five years ago the railroads in many states practically dominated the politics of those states when it came to any legislation in which it might be interested, and anyone who is only half-way acquainted with the United States knows that, if there is one thing in this country we will not stand, it is the domination of any particular interest, whether it be the interest of capital or the interest of labor. (Applause.)

The Interstate Commerce Commission was formed as a political measure in order to curb the railroads. It was realized that whoever controlled the commerce of the country, with proper restrictions, was very apt to control its politics. Yet when you ship your goods through Chicago to the port of Baltimore and on to South America, you control, for example, until the goods get to Baltimore, and then you turn them loose, as it were, until they get to South America, and if your ships under your own guidance do not handle them to a reasonable extent, it follows that whoever does handle them has a power over your politics and over your foreign trade, which, no matter how beneficently used, cannot be maintained indefinitely. It seems to me that although we are willing, and more than willing, that all countries should be joined together in advancing the welfare of the world—and we know, for instance, in this great business of cleansing the world, and the carrying trade and commerce, some nations excel—it does seem to me that we must have a control over a part of it, a reasonable part of it which originates with us, and which goes to our own particular customer.

Now, not only is building involved, not only is the question of money involved in the operation of ships, but the great questions of insurance and classification are involved. I have been accused somewhat lately of going after my good friends from the other side, who insure and also classify ships, and I would have them understand it is only in the most friendly spirit of competition we approach that subject, but we do believe that there are many vital parts in the building of a great marine, and we do believe that we should have here an insurance for that marine, not for all of it, maybe, but for a large part of it; that we should be able to have an American classification society which will rank with the best, and I want to make it clear that I am heartily in favor of it, just as I am in favor of building the ships—as many of them as we can build—in our own yards. (Applause.)

In the case of any body of men or of any society whatever, gentlemen, we know that competition is a pretty good thing. We are trying to get rid of competition, more or less, and people who do not think very soundly from an ethical standpoint are trying to get rid of it altogether. In fact, if things become too large and we have no competition, they become topheavy, and it is good for all of us that we have to go into the market and bid on things, and get down on our marrow bones, once in a while, and work. These times of shipbuilding are so nice and pleasant that it is hard to think of the days, when a little freighter was to be let in the port of New York, it would bring us flocking to the men who were to build the ship, with our figures all made out, to see who could build it and lose the most money on it. (Applause.)

Of course we prospered under government orders and under the Macy Board Award, and we have got our business up on the roof, as it were. We have got it so that ships that were commonly built for $60 a ton now cost from $180 to $200. Maybe we are going to build up a great marine on that basis. Gentlemen, I do not think so. I do not think that American shipbuilding will become a permanent institution until it has gotten a little nearer to bedrock than $180 a ton. (Applause.) Make them standardized or any other kind—one of the difficulties with the job is when you get the price down and so that you can build them and turn them out like hot cakes, you have too many of that kind. Of course the cargo ship is important, but a more important type of ship is the great ships that carry the mail and passengers and the fast freight—the liners. From the shipbuilding standpoint, from the naval architectural standpoint, these are the ships that will be developed in the future in this country to an enormous extent, if we have any marine. However, we must have something to practice on to work up to that. We ought not to be asked to build a 1,000-foot ship, going at 30 knots an hour. I do not mind telling you I do not know how to build a 1,000-foot ship which will go 30 knots an hour, of the merchant type, but I do believe we could learn how to work up to it. People who think we cannot design and build ships of the highest type in this country need but look over the latest of our naval vessels. For a good many years our friends on the other side of the water said that our naval ships were built so lightly that they would not hold together. I remember that was said in the case of the old Iowa, I think. We are building a new Iowa now, of about 43,000 tons, and the framing is not much heavier than the old Iowa. In naval design we have not only designed as good ships as are built in any country, but in some particulars probably a little better, and in that connection I cannot talk about ship designing or shipbuilding without telling—a lot of you know—without telling the new members of this Society and the visitors that probably the man most responsible for that (and I am sure on the other side they would agree with this), probably the most distinguished ship designer in the world, is sitting at the speakers' table now—he will probably go after me for saying this—but you will find that his works, what he has written, what he has planned and what he has done are honored and followed in ship design, even throughout Europe. I refer to Admiral D. W. Taylor. (Loud applause.)

Not only has he done great design work in naval vessels, but his advice and assistance in the designing and building of merchant ships have been of great value.

Sometimes men say they think there is too much navy in this shipbuilding game (cries of "Hear, hear"), and the reason is perfectly apparent. Before the war, the Navy was all the shipbuilding we had upon which to concentrate our talents. The Naval Academy turned out men who were sent abroad to learn shipbuilding and learn designing, who were almost the only men given a thorough education in ship design and engine design, and were, therefore, the most easily prepared of any to take up this work. Now that we have the Massachusetts Institute of Technology and other splendid places, more and more men are being educated in this country along these lines, so that not only in ship design and shipbuilding, but in the arts of the sea, we are coming back into our own. (Applause.)

I was at a school at Hampton Roads the other day where about 400 young men are being brought in from the farms to learn to be seamen, and they are going to make good seamen, as we found out during the war. I talked to some of the boys as to why they came and where they were going, and they said they had concluded to adopt a career on the sea because they had the instinct of the sea. Our forefathers came to this country because they believed in reaching out into a new country and going out into trade and commerce and seeing the world, and whenever that instinct of romance and desire to move about is destroyed in our people, we had better stop. When boys no longer want to go to sea and go out over the earth and see new lands, then the spirit of adventure will be dead. We have filled up the United States pretty well now, and our boys are eager to go out on the ocean again. It will take a generation of effort in order to put us on the sea as we really belong, but we are getting there, and our boys will go to sea in good American ships and will become American officers, and we will have our American seamen, and as our friends on the other side continue their development, we will go with them hand in hand to the ends of the world and carry the message of our civilization. (Applause.)

The President:—Mr. Ferguson's remarks leave nothing for the president to say. We knew exactly the high character of what was coming from him when he arose. The next and last toast is "The Engineer in the Navy." Those who were present at the technical meeting yesterday afternoon heard of the exceptionally fine work performed, while he was on duty at the New York Navy Yard, by the one who responds to this toast. I will say no more, but present to you Captain Earl P. Jessop, U. S. Navy, who will speak for "The Engineer in the Navy." (Applause.)

ADDRESS BY CAPTAIN EARL P. JESSOP.

Mr. President, ladies and gentlemen, I have not a megaphone, and I do not know whether I am going to be able to make myself heard. I reached the shores of the United States on Monday last, after having been out of the country for eighteen months, and on board ship I received a radio stating that I was billed to speak before this august assemblage tonight. They said nothing about the subject, so when I got in I telephoned and was told the subject was to be:—"The Engineer in the Navy." That is sufficiently comprehensive, but in order to reduce it to something like talking proportions, I am going to add to it a text. The text of my remarks will be, "And the rabbit climbed a tree," taken from Uncle Remus, 14th Chapter and 5th Verse. You remember that Uncle Remus told a story of the time the fox chased the rabbit and got the rabbit in such a position that he found it impossible to save it, and so he said, "That rabbit climbed a tree." The little boy said, "Uncle Remus, rabbits cannot climb trees." Uncle Remus said, "No, honey, I know they cannot climb a tree, but that rabbit sure had to climb the tree," and that has been the experience of the engineer in the Navy.

They amalgamated us in 1899—they said the line must be engineers and the engineers must be the line. Now there are a great many engineers in the United States who believe that that order was a mistake. I wish to say tonight that there never was a better order issued in the United States Navy. The ships prior to that time did not run—that was not due to the personnel, but it was absolutely due to the system. The men on deck knew nothing about below decks, and the men below decks knew nothing, and did not care a damn, about what went on up on deck. The thing the engineer found when he got on deck was that he had to spend all his time hiding behind ventilators getting away from the captain's wrath. We had been used to that on deck and it did not bother us; when we went down below we found machines we never dreamed of, and in order to show you how it worked I will tell you of my own experience. I do not know the experience of anybody else so well, and the relation of my own may sound egotistical, because I may use the personal pronoun "I" quite frequently, but it is the only way to let you know how an engineer in the Navy grows up. I am not speaking of the old designers, of the men trained as engineers from the start, of the men who recognized Mr. Thermodynamics when they met him on the street, but I am talking of the operative engineer today in the United States Navy.

I am not an engineer as you gentlemen recognize the term. I am merely an operating engineer and have got what knowledge I possess by practice rather than theory. My first engineer experience of any magnitude was in 1905, and it was absolutely deliberate. I had spent some nine years outside the Naval Academy, and I decided of my own motion that it was high time that I knew something about what went on down below. I went to the Navy

Department and deliberately asked them to assign me to the Milwaukee, which was then building on the west coast, and at that time was about to have its machinery fitted. I spent one year in seeing the engines go into the ship and the connections being made, and that was my entire experience during that period of time. Everybody said I was crazy, but I figured it this way: If they would let me go to the Union Iron Works, and spend a year watching the piping going in, the valves going in, the engines going in and Mr. Peabody's boilers going in, at the end of that time I ought to have sense enough to observe enough to take the personnel they gave me and use the brains of those fellows to run that machinery. That is the way I figured it.

Prior to putting the ship in commission I had to organize the engineering department, and knew absolutely nothing about engineering organization. How did I do it? I had seven warrant machinists, men who had spent their lives among machinery plants. I asked them to make recommendations, and I was very fortunate. One man brought in a brown sheet of paper with a pencil diagram, showing all the stations of the complete organization. That organization was not all that it should have been, but it was a foundation on which I could start. I took that brown sheet of paper and worked over it for three months. My personnel was to consist of 250 men—it was large for those days. The ship was 25,000 horse-power and had sixteen Babcock and Wilcox boilers. By the time I got through with this three months' study of the plan, I took what I considered the finished product and put it up to seven or eight engineers and asked them to criticise it. They did criticise it, and after they had got through with the criticism, I put the plan into effect—I used some of the criticism and some of it I did not, as I thought I understood the matter a good deal better than some of those engineers did.

I went to sea in that ship. At that time we did not know very much about taking care of watertube boilers—we do not know too much about it today, as the Shipping Board is finding out—but I had seen an article in one of the service magazines about taking care of boiler water with chemicals. I did not know anything about taking care of boiler water. I did not think you had to look out for the water, but the metal, and did not think the water had anything to do with it. I read this article, and it seemed like good, sound sense. I saw the warrant officer having charge of the firerooms and told him that we would test the water chemically. He thought I was crazy—he knew I did not know any engineering, but I insisted. We had salinometers and I used them for a week. At the end of the week I was in the fireroom, and the men were taking salinometer readings. The salinometer showed that the water was fresh, but I tasted it and the salt in it nearly knocked me down. I took a small bucket and put some chemically pure water in it. This bucket I placed inside a larger bucket in which there was some other water. I set the larger bucket over a fire and brought the water in the inner bucket up to 89.5° C. I then placed five of the hygrometers which were furnished with the salinometers in the water in the inner bucket and they gave me five different readings. One said the water was 4/32 salt, another 3/32, another 2/32, another said it was fresh, and the fifth said it was alcohol. I did not have to be an engineer to decide that something was wrong with the salinometers, so I took them off the boilers, put them in the storeroom, and reported to the Navy Department what I had done. They did not even answer my letter.

We went to sea, and the first day out the water got up to 700 grains of salt per gallon. I was frightened. I was afraid the salt would chew off the connections inside. But I had a nine-day run, and did not have sense enough to tell the skipper to turn back. I stuck to my chemicals, and I put alkaline substances into the water, to keep it alkali—I put enough in to be sure that I had alkaline water all the time, and took the tests about once an hour. We ran for nine days, and we ended up with 1,700 grains per gallon in the water in the Babcock and Wilcox boilers, but when I opened them up they were as clean as a whistle. There was no sign of corrosion; they were perfectly clean, beautiful boilers.

We made the return trip under the same conditions and with the same experience. When we got back the boilers were as clean as if no salt had gone into them, and as if they had not been full of sea water, not all the time, but a great deal of the time.

I was interested to know what could be done in case condensers were not tight, so I took one boiler and ran a test as follows: I lighted fires in it and ran sufficient salt water in it to make a solution which had 400 grains of salt per gallon. I gradually brought the salinity up to 800 grains per gallon and kept the steam on the boiler for four months with the water in that condition. At the end of that time we let fires die out and opened up the boiler. The upper part of the boiler showed no signs of having had salt in the water, the steam drum and the upper tube being clean and in fine condition. It was not until we got down to the three lower rows of tubes that we found any deposit, and this deposit was not hard scale but was a light, fluffy deposit which crumbled when touched and was very easily washed out.

I do not imply that salt water is a good thing to have in a boiler, but with the present development of the condenser, sooner or later engineers will have to run for short intervals with salty water, and when they do they should know what to do to prevent harm to the boilers. I had discovered this because I went at the problem with no preconceived notions to prevent me getting to the bottom of the problem. After that test was over I knew I could take care of my boilers no matter what happened, and I made the statement in a letter to the Bureau of Steam Engineering that salt water would not hurt a boiler if proper precautions were taken. Any engineer who knows how to chemically test water and carefully maintains a slightly alkaline condition in his boilers need have no fear of salt hurting his boilers on any run he may have to make. It is of course better to have no salt in the water, but, as I have just said, salt will get in sometimes and the ostrich method of paying no attention to it and trusting to luck will not work.

I lost one tube, and that tube was lost within a month of the time when it was put into the ship, because the tube was defective from the start, but in three years there was no other tube which showed, by micrometer, that it had lost any material, and the chief engineer said the tubes were as good as when they went into commission. Similar vessels all around were losing tubes right and left. Why? Because I started in, not knowing a thing, and knowing I did not know, and I happened to see the printed article to which I have referred. I adopted it as a Bible, and it happened to be right. You know an expert is described as a dangerous man. A man may be an expert without knowing it, but the minute he begins to be an expert and knows it, fight shy of him—he is a dangerous man. He usually is in such a groove that he cannot see over the top, and he will mislead you—unintentionally, of course.

The engineering experience that I have had has taught me that the big thing is to use the other fellow's brains. A man once came to me and said, "I heard one of your boys say a nasty thing about you. He said you were a fellow who made a great reputation out of other people's brains." I replied, "That is not nasty. It is a compliment."

The next move from the Milwaukee was to the Arkansas. I had never seen a turbine. I went to the New York Shipbuilding Company and saw these things going in, the piping, the boilers and the turbine, and I doped out this thing about turbines—keep your bearings up, keep the oil going, keep your clearances properly and the rest will run itself. I finished that job of two and a half years and never lost a tube or any material, and the thing ran beautifully. Why? Because I followed my little Bible.

From the Arkansas I went to the Eastern Mediterranean, and Admiral Burd ordered me back as chief engineer of the New York yard. Admiral Burd never knew how much engineering I knew. I had been his first assistant on the old Philadelphia and did not know very much about engineering; all I did was to keep my mouth shut, and the admiral got the idea that I was a good engineer. He ordered me to the New York yard as chief engineer. I got there and found a wonderful organization, left by my friend, Louis C. Richardson. I did not issue many orders, and what helped me to save the situation in that case was the brains they gave me in the personnel. The only thing I have done as an engineer is to use the other fellow's brains, and to have sense enough to know which fellow had the right kind of brains.

One of the great secrets in organization is to put the right fellow in the right place. I have followed two rules in organization and management and they are these:—Authority engenders responsibility; responsibility presupposes authority. You would be surprised if you looked over this country to find how many big corporations break those two rules and are not satisfactory corporations for that reason. You would be surprised to find how many general managers go into their shops and go to workmen and tell them to do this and that, and pass by the foreman or master mechanic, and you will not be surprised to know that these men are not successful managers. The only thing they have to do in order to become successful is to go back and put into effect these two rules I put into practice in the case of the Milwaukee and in the Arkansas, and that was the reason I was more or less successful, not that I knew engineering, because I did not, but I did know how to use the brains and experience of the fellow that did know.

In connection with that, I want to talk about amalgamation. There has been an effort in the last few years to put back the Engineering Corps of the Navy. Perhaps the people who advanced that proposition did not appreciate what they were doing, but I can tell them that they are doing an iniquitous thing. The reason it was necessary to amalgamate was not that the line officers were going to teach the engineers proper engineering, but it is a fact, gentlemen, that a ship is a unit, and you cannot have two organizations in one ship and work together as a unit. The corps system on the operative side will not make a unit, and we are trying to form a new Engineering Corps in the Navy—those fellows are on the beach, not going to sea. Why? Because they have become commanders, they do not have to go to sea, they lose touch with the engineers they have charge of, and they sit back in Washington and tell us what we can do.

It is the same old story coming up again, and it will end in putting the Navy back again to the place where the captain goes to the speaking tube and yells an order down. The man down below does not understand the order, so the captain says, "Who is the damn fool at the end of this tube?" and the boy says, "Not at this end, sir." That is what you are coming to if you put the new Engineering Corps into effect in the Navy; we do not need it. There are enough youngsters in the Navy to do the designing work and keep the designing work where it needs to be, and there are enough of you gentlemen in civil life who are perfectly willing to design anything we want, but ours is an operative proposition, and you cannot operate with two corps on board the ship, attempting to do different things at the same time.

While at the New York Yard I ran into German shipwork, as some of you may know. I understand that Mr. Anderson told you about it yesterday. There are some things about that German shipwork you ought to know and do not. In the first place, the German shipwork was a distinct illustration of my text, "The rabbit had to climb the tree." These ships were the big part of our transportation facilities. I think, outside of the German ships at that time, we could transport 20,000 men a month, so we had to take them. The estimate was that it would take eighteen months to get them ready for sea. That looked all right, but we could not take so long. There were men in Washington who were saying that Europe was done if we could not get men abroad quickly, and that proved to be absolutely true.

In order to get these ships we turned to electric welding. It was a new departure in marine engineering to apply electric welding to cast-iron cylinders the way we did. We started the work. We expected to have some failures. We certainly did not expect to have the wonderful success we did make out of it. I believe that we had more luck than is generally understood, but it did offer a quick solution to a slow problem; the result was that in five months we had turned out all the ships which were estimated to take eighteen months. The first ship which was finished, on which the estimate was ten months, was turned out in three weeks. You could not get around it.

During that work we found some very curious things. One of the side issues was that we found $2,000,000 worth of machine tools on a Hoboken pier. We needed machine tools very badly at that time. This discovery was reported by a Customs House man. I went to the Collector of Customs and asked him to put a stop on the shipment of these tools. He said he could not do it—that they were going to neutrals, shipped by neutrals and he could not touch them. I said:—"The United States is at war, and you have no right to let these things go out of the United States unless you are absolutely sure that they are not going to Germany, and you do not know that to be the case." He said:—"I will have to make you responsible for it." I replied:—"My second name is responsibility." He telegraphed to Washington and had the shipment of the tools stopped. There is something about red tape which perhaps you do not know. You can work it backward. After we put a stop on the shipment, it took us three months to find out how to get it off so that we could use the tools. At the time that we opened the consignment of tools we found that they were all destined for Germany. That was a side issue.

Another one was in connection with the Leviathan, when the question of cleaning her bottom came up. I knew she had been lying there in the North River for three years or more, and I expected that her speed could not be greater than 17 knots. We did not think that was enough. I called up the manager of the Chapman Merrit Wrecking Company, and said:—"Have you divers who can clean the bottom of the Leviathan where she lies?" We talked about the great necessity of getting the ship. I aroused his enthusiasm, and Monday morning he got six divers on the work. It took them six weeks to do the work, and at the end of that time we inspected her, found she was clean, and she went to sea three months later, made 22.5 knots, went abroad, and when she was put in the dock she was as clean as a whistle. (Applause.) I had learned something about the North River. I had learned of a ship that had left dry dock for two months, and at the end of that time she had a complete outer layer of barnacles. We put her in the dock. When the scrapers were put on, we found the first coating was a coating of slime, and the barnacles adhered to that; the paint was not attacked, and when we took off the barnacles the paint was just as good as new. Many people thought I was ruining the bottom of the Leviathan, without a chance to paint her, but I happened to know that about the North River. My judgment proved to be correct, as her paint was in good condition when she went to Europe.

I want to tell you a little more about how much further we have gone in welding since that time. The latest exhibition of welding of any magnitude that I know anything about was in the case of the Oklahoma. She came to Brest last summer to bring the President back. The President made several false starts and in one of these false starts the Oklahoma warmed her engines up, got a slug of water in the high-pressure cylinder, and blew the high-pressure cylinder head off. Three-quarters of the flange of that cast-iron cylinder head was cracked, and two feet of the circumference broken square out with the strengthening ribs, and every strengthening rib in the head was cracked. The head weighed 2,200 pounds and my casting capacity on the Bridgeport was 2,000 pounds, so we were up against it. Two alternatives confronted us—one was to construct a pattern to make a new casting, and the other was to electric-weld this cast-iron head. We cast a new head, but it was defective, because it was beyond the capacity of my furnace. Then we electric-welded the head and the ship came home with the President with the cast-iron head welded—welded in seventeen places. (Applause.)

Anybody who thinks that welding is not to be depended upon had better get off the track, because welding is going to take a lot of your troubles off your hands. You do not believe it, probably, but there is no question about it, with proper welding equipment (and you can get it in the United States better than anywhere else in the world) and with proper supervision, you can use welding in the strength members of vessels with perfect safety.

There is one ship going to sea in the United States Navy today with 85 frames electrically welded. She was smashed abroad, and that was the only way to get the repairs done. The first thing she did was to get in a gale and get a terrible battering, and not a sign of any weakness showed up in the welding.

My next job was the Bridgeport, the repair ship. I took her to France. A German submarine fired a shot at her, but the torpedo missed her by a narrow margin. When we arrived there the work of repairing torpedo boats went on, and after that we took over the care of the cargo steamers (you know that all the atrocities were not committed by the Germans in Belgium and France; some of them were committed in shipyards in the United States). Some of the shipyards used palpitating bulkheads in the ships, and if they got an especially weak bulkhead, they loved to put the auxiliary machinery on the bulkhead, so that the bulkhead would work and the machinery would not. Some of them used a curve of sines as bedplates for engines. In the case of one ship, I had to bore each bearing to a different offset and then make drawings of these bearings and their offsets, so that the next time they put in a bearing they would know how to place it. We ripped the turbines out of the ship, realigned shafting, put the turbines back and sent the ship home.

The Murray had 70 per cent of her keel carried away. When she came in, we rebuilt the hull and sent her home.

The great advantage about the work over there was the fact that you were absolutely left alone. There was nobody to say "You cannot do this" or "You cannot do that." We had to get the ships back. We were allowed to lay our own plans and go ahead and finish them, and the day the Bridgeport left the port of Brest there was not a tug or lighter or ship that was not fully repaired, and everything that carried the United States flag was out of Brest three days after we left—we did a complete job. (Applause.)

Just a word more. I would like to refer to a matter that the president spoke of earlier in the evening. It impressed itself on my mind all the time I was in Europe. It was this—that we stand to get nothing out of the war except a higher conception of governmental affairs; but if we get that, if the individual citizen knows and appreciates his responsibilities for his government and the way it is run; if he appreciates the fact that going to the polls once in four years is not the full service that the citizen owes his government; that if he appreciates, when any law is passed, that he is a part in this government and becomes responsible for that law, and therefore it is his duty to take an interest in seeing that the law is all that it should be, the day that the ducated and trained minds in this country take such a vital interest in the governmental functions that nothing can be done without their influence to make that thing right and the best for the Government, not the best for any small coterie of people, but the best for the people as a whole, the country will have gotten the full effect of the war and it will have been worth all it cost, and the men who lie in the fields of France and Flanders will not have died in vain. (Applause.)

THE PRESIDENT:—Evidently the Engineer in the Navy is all right, and we owe our thanks to Captain Jessup for his splendid tribute.

We have had a most interesting evening, gentlemen, and, in closing, it seems only proper to record our appreciation of the assistance given by all who have participated in our meetings. In this connection, I desire, on behalf of the Society, to convey our especial thanks to Messrs. C. M. Wales, W. H. Todd, J. H. Gardner, E. H. Peabody, F. P. Palen, and Commander Stevenson Taylor, who served so effectively as members of our Entertainment Committee. The thanks of the Society are also particularly due to our most efficient secretary, Mr. Cox, and his associates.

The meeting is now adjourned without day.

Deaths, 1919

ALBERT ALLEN

MEMBER

Albert Allen was born January 10, 1873. He served an apprenticeship in the shipyard of Furness, Withy & Company, Ltd., West Hartlepool, England, and was employed by them until 1905. He afterwards entered the employ of Lloyd's Register of Shipping at Baltimore and New York. His principal duties covered the inspection, during construction, of vessels building at the above-named ports for classification with that society.

He died March 2, 1919.

EDWARD PAYSON BATES

MEMBER

Edward Payson Bates was born March 3, 1844, in Savannah, Ga. At the age of seventeen he entered the machine shop of Hobart B. Bigelow, in New Haven, Conn., who was afterwards governor of the State of Connecticut. He remained there only a short time when he moved to Albany, N. Y., and entered a machine shop. He later moved to New York City, where he was granted a marine license and for several years went to sea as an engineer, seeing service on a transport bearing wounded from Libby Prison.

Soon after this he entered business with Mr. Willis Warner at Syracuse, N. Y., in the erection of steam-heating apparatus. After the death of Mr. Warner he established a business under the firm name of Bates & Johnson Co., locating branches in eleven cities. The business was later incorporated, and Mr. Bates was president at the time of his death.

Mr. Bates was a member of this Society, charter and life member of the Technology Club of Syracuse, life member of the Mayflower Society of Mass., life director of the American Bible Society, life member of the Archeological Institute of America, member of the Robert Fulton Memorial As-

sociation, life member of the Bates Association, and director of the Syracuse Museum of the Fine Arts. In 1910, Mr. Bates was appointed official delegate of this Society to the Fiftieth Anniversary of the foundation of the Institution of Naval Architects, which was held in London.

He died August 3, 1919.

MARSHALL TEN BROECK DAVIDSON

MEMBER

Mr. Davidson was born in Albany, N. Y., 1837, and from his early boyhood showed great interest in steam machinery, particularly the locomotive and steamboat engines. In 1857 he entered a marine engine machine shop, Ft. North Moore Street, New York City, and later went to sea as assistant engineer on ocean steamers.

Upon the breaking out of the Civil War, he was assistant engineer on a steamer running from San Francisco to the various settlements on Puget Sound. He came east in 1862 as volunteer, second assistant engineer in the Naval Service, but was requested to accept a position as chief engineer of an army transport building at Wilmington, Del. He superintended the construction and installation of the machinery of this vessel, and after its completion received a commission as chief engineer in the Revenue Cutter Service, superintending the building and installation of the machinery for that service.

At the close of the war, Mr. Davidson became a contracting engineer, and in 1878 started the business which is now carried on in Brooklyn, N. Y., under the name of the M. T. Davidson Co., builders of pumps and pumping engines.

His knowledge of character and strong attachments to those he liked and trusted created the faithful staff whom he gradually brought to operate the business on the lines laid down by him, and in the corporation formed some years ago to carry on his work, the same "Davidson" men who had been with him for twenty-five years or more were given charge of the business to carry on the principles on which they are so well grounded.

Mr. Davidson was closely identified with the progress of engineering in the United States Navy since the building of the "White Fleet," which marked the beginning of our modern warship construction. His practical knowledge of engineering, especially as applied to pumping machinery,

placed him at the head of his profession in that line, and he was identified with the work of the Mississippi and Missouri River Commissions of the United States Government, large pumping engine installations in Brooklyn, N. Y., and many other cities. His favorite study was marine work, and the development of pumps as now built for use on shipboard was largely due to his inventive genius and exact knowledge of the requirements.

Mr. Davidson was a member of this Society from its foundation, one of the oldest members of the American Society of Mechanical Engineers, also of the Naval Order of the United States, Associate Society U. S. Grant Post No. 327, The Columbia County Association, Life Member of the Navy League, and was, from its conception, an active and life member of the late Union League Club of Brooklyn.

Mr. Davidson died on April 10, 1919.

WILLIAM D. DICKEY

MEMBER

Mr. William D. Dickey was born July 16, 1852, in Belfast, Ireland, of Scotch parentage.

Before coming to this country he was apprenticed to a firm of shipbuilders in Belfast. On coming to this country in 1870, he joined the firm of Handren & Ripley as superintendent. This firm eventually became the firm of Handren & Robins and later the Erie Basin Dry Docks, later still the John N. Robins Co. He remained with this firm until 1905, when he established the Shooter's Island Shipyard, with which he was connected until 1909. From 1913 to 1915 he was manager for James Shewan & Sons, and following this he was connected with the United States Shipping Board in an advisory capacity.

His education consisted of a course in high school in Belfast and a four-year course in mechanical engineering at Cooper Institute in New York City.

In 1873 he married Rosa Mulholland, the daughter of the Rev. J. R. Mulholland, of Owen Sound, Canada, who with his son, W. E. Dickey, survives him. Mr. Dickey was a member of this Society since its foundation.

He died on June 7, 1919.

CLEMENT ACTON GRISCOM

ASSOCIATE

Clement Acton Griscom was born in Philadelphia, Pa., on June 20, 1868. He was the eldest son of the first president of the Society of Naval Architects and Marine Engineers.

Educated abroad and at the University of Pennsylvania (Ph.B., 1887), Mr. Griscom became supervisor and then manager of the International Navigation Company, and general manager of the International Mercantile Marine Company. In these positions he accomplished much for American shipping and marine development, being responsible for the organization and for many of the improvements which at that time helped to place the ships of the American and Red Star Lines in the forefront of the transatlantic trade.

He resigned this position in 1904 to devote his energies to the management of The James Reilly Repair and Supply Company, of which he was president. This company became later The Griscom-Spencer Company, and finally the Griscom-Russell Company, designers and manufacturers of marine auxiliaries which have become the standard of the American Navy and shipping world. His wide knowledge of the industry enabled him to foresee the great marine expansion which now has become a reality.

By selecting the best engineering talent he could find; by large expenditure of money, and, above all, by means of his own tireless energy, perseverance and faith, he was able to produce and perfect marine auxiliaries which, during the great war, were largely used in vessels of the American Navy and Shipping Board.

Mr. Griscom died December 30, 1918.

FIRST LIEUTENANT PARR HOOPER, R. A. F.

JUNIOR

Lieutenant Hooper was born in Baltimore, September 5, 1892, the son of Mr. Herbert Hooper and a descendant of one of the old shipowning familes of that port. After passing through the Baltimore Polytechnic Institute and Sibley College, Cornell, as a mechanical engineer, he was for some time with the Lanston Monotype Co. in Philadelphia, leaving that em-

ployment to enter the yard of the New York Shipbuilding Co. at Camden, N. J. Here he distinguished himself by an indefatigable application to practical problems of construction and an enthusiastic study of timekeeping and payment systems.

The entry of the United States into the war broke in upon this; and Hooper, who had, from his Cornell days, looked with deep interest on the growing science of aviation, and stirred, moreover, by a love of romance and adventure, which was a strong element in his character, enlisted, early in 1917, in the Aviation Section of the U. S. Signal Corps. He passed through the School of Military Aeronautics at the Ohio State University. Later he was assigned to the 32d Squadron of British Expeditionary Forces, and, after further training in England, he was engaged on the British front, where he lost his life.

On June 10, 1918, near Sorrel Chateau, south of Lassigny, about 11 o'clock in the morning, while leading an aerial patrol, First Lieut. Parr Hooper crashed down into the German lines. It is impossible to say whether he was hit, but it is probable he was killed by the fall. Diligent inquiry, covering many months, has failed to elicit any details of his fate.

Maj. J. C. Russell, of the Royal Air Forces, writes:—

"He proved himself an exceedingly brave and good leader. He will be a great loss to the Flying Corps, the U. S. Flying Corps, and especially to this squadron at the present time. He would have been with me only a few weeks longer as I should have sent him as a Flight Commander to the U. S. Flying Corps."

Lieutenant Hooper was of an engaging personality, self-reliant but devoid of assumption, moral, affable and deservedly popular with his associates.

Thus passed in the glowing enthusiasm of his young manhood the only member of our Society who died in action in the Great War, and "sealed with his blood his heart's desire."

RICHARD LANO NEWMAN

MEMBER

Mr. Newman was born at Dorsetshire, England, July 15, 1864. He entered the British Navy at thirteen and a half years of age as a midshipman, but four years later, persuaded by his uncle, a high naval officer, that a greater future awaited young men of education especially along the line of

shipbuilders, he left the Navy and studied at what is now known as London University.

After completing his course of naval architecture and engineering under the tutelage of John Penn and Sons, at the age of twenty-one years he became draughtsman for the Earles Shipbuilding Company, and later joined Handslay Sons & Field of London, who at that time had a ten-year contract to design ships for Italy and Spain.

After finishing with the Handslay firm, he came to the United States, where he met Charles Cramp at Philadelphia. Mr. Cramp was attracted to the young English shipbuilder and offered him a position with his organization, which was then doing a great deal of work for the Government.

During the four years that Mr. Newman was with Cramp's yard, many battleships were built there and also the St. Louis and St. Paul.

Leaving Philadelphia, Mr. Newman went to Cleveland, Ohio, with the Globe Iron Works. During his management amalgamation of the ten largest lakes shipyards was consummated, known as the American Shipbuilding Company .

Later Mr. Newman joined the late H. G. Morse in building the New York Shipbuilding Plant at Camden, N. J.

Eleven years later he established offices in Montreal, Canada, as consulting engineer and did much work for the Navy Department of the Canadian Government.

When the York River Shipbuilding Corporation was organized for the building of United States Shipping Board Ferris type vessels, the directors of the company secured the services of Mr. Newman to build and operate the plant at West Point, Va., in the building of government vessels, in which work he was actively engaged, being president and general manager of the York River Shipbuilding Corporation at the time of his death.

Mr. Newman is survived by his wife and two daughters.

Mr. Newman died January 26, 1919.

WILLIAM H. PLEASANTS

ASSOCIATE

Mr. Pleasants was born in Richmond, Va., April 29, 1863, and began his business career in that city as a clerk with the Richmond & Danville Railway, some years later going to Jacksonville, Fla., where he was general

freight agent of the Florida Central & Peninsula Railroad, now the Seaboard Air Line Railroad. In 1890 he became freight and passenger agent of the Ocean Steamship Company in Savannah, and after a period in this post he returned to the Seaboard Air Line, but it was not long before he again joined the Ocean Steamship Company as vice-president and general manager, at that time taking up his residence in New York City, where he had since resided. At a meeting of the directors in Savannah on April 14, 1915, he was elected president of the Ocean Steamship Company.

Mr. Pleasants was recognized as one of the leading steamship transportation men in the country, and he had a prominent part in the development of the Ocean Steamship Company. In February, 1918, Secretary of the Treasury W. G. McAdoo sent for Mr. Pleasants and asked him to become manager of the Marine Section of the United States Railroad Administration, which Mr. Pleasants agreed to undertake, although his health was then failing. After a few months he resigned on account of his continued ill-health.

Surviving Mr. Pleasants are his mother, Mrs. J. P. Pleasants of Richmond, Va., and a brother, Charles M. Pleasants, also of Richmond.

RICHARD C. VEIT

ASSOCIATE

Richard C. Veit was born in Manhattan, N. Y., November 17, 1855. Early in life he entered the employ of Rockefeller, Andrews & Flagler, subsequently the Standard Oil Company, with which interests he was identified for fifty-two years.

In 1880 he was placed in charge of the Lighterage Department of the Standard Oil Company of New York, and great advancement was made by this department under his direction.

For many years Mr. Veit was a director of the Standard Oil Company of New York. In 1911 he became secretary.

He was interested in many philanthropic movements and was identified with the old J. Hood Wright Memorial Hospital, the American Museum of Natural History, the Metropolitan Museum of Art, and the New York Zoological Society.

Mr. Veit had a summer home at Sea Gate, Brooklyn, N. Y., where he

spent much of his time. He was a member of the Atlantic Yacht Club at Sea Gate and of the New York Yacht Club, also a member and governor of the Lotos Club and a member of the Maritime Exchange.

He is survived by his wife and two sons, Russell C. and Kenneth A.

He died in Yokohama, Japan, on August 29, 1919, while on an extended tour of the Orient.

RUDOLPH VEITH

MEMBER

JOHN M. WILLIAMSON

MEMBER

John M. Williamson was born February 28, 1885. He served six years in a shipyard engine shop and drawing office. He was chief engineer and later marine superintendent for W. R. Grace and Co., New York, for four years.

He was afterwards employed as a surveyor by the American Bureau of Shipping and still later as engineer surveyor for the Lloyd's Register of Shipping. His principal duties covered the inspection and test of forgings, castings, etc., intended for vessels classed or to be classed with Lloyd's. His resignation was accepted by Lloyd's on April 21, 1919. He died in May, 1919.

PLATES

To illustrate paper on "Methods Employed in the Construction of Concrete Ships," by R. J. Wig, Esq., Visitor.

TABLE 1

CHARACTERISTIC DATA ON CONCRETE SHIPS

TYPE OF SHIP DESIGN NUMBER OR NAME	CARGO ATLANTUS	CARGO NO. 2	CARGO POLIAS	CARGO NO. 69	TANKER NO. 70
Length B P	250-0	268-0	268-0	420-0	420-0
Length O A	260- 2 1/2	282-2	281-9	434-3	434-3
Beam (Max)	43-6	46-0	46-0	54-0	54-0
Draft, Full Cargo	22-6	24-0	23-6	26-0	26-0
Depth	26-9	28-3	26-6	36-0	36-0
Displacement, Full Cargo	5240	6310	6220	13000	13000
Deadweight	2766	3590	3347	7499	7445
Speed in Knots	10 1/2	10 1/2	10 1/2	10 1/2	10 1/2
Metacentric Height (Cargo-Light)	:	2.20 : 3.45	:	2.27 : 3.20	2.60 : 3.30
Frame Spacing	5'-0"	5'-0"	4'-0"	12'-9"	4'-3"
Shell Thickness - Bottom	5"	5"	5"	4 3/4	5
" " - Side	5" & 5-1/2"	4"	5"	6 1/2	4
Type of Deck Erections - Poop	Concrete	Concrete	Concrete	Wood	Wood
" " " " Bridge	Concrete	Concrete	Concrete	Concrete	Concrete
" " " " Forecastle	Concrete	Concrete	Concrete	Concrete	Concrete
Reinforcing Bars - Wt. in Tons.	500	506		1540	1550
Concrete, Volume in Cubic yds.	1092	1120	1000	2550	2660
Type Engine	Triple Exp.	Triple Exp	Triple Exp	Triple Exp	Triple Exp
Developed I.H.P.	1520	1520	1520	2800	2800
Developed RPM	94	94	94		
Boiler	Water Tube	Water Tube	Water Tube	Water tube	Water Tube
Fuel	Oil	Oil	Coal	Oil	Oil

NORRIS PETERS, INC., LITHO., WASHINGTON, D. C.

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TABLE 2.

TYPICAL STEEL ANALYSIS

		Physical Analysis				Chemical Analysis				
Steel Rolled By	For use at	Elastic Limit	Ultimate Strength	% Elongat'n in 8"	% Reduct'n in Area	Carbon	Manganese	Phos.	Sulphur	Grade
Atlantic Steel Company	Wilmington	42,930	62,030	28.1	49.1	.18	.43	.042	.057	Structural
Llewellyn Iron Works	Oakland San Diego	35,960	57,020	29.2		.20	.43	.018	.043	Structural
Tenn. Coal, Iron & R. R. Co.	Mobile	62,230	97,356	16.9	34.2	.42 .52	.50 .90	.029	.03-.06	Shell Discard
Bethlehem Steel Company	Jacksonville	54,550	97,345	16.0	24.0	.50	.77	.035	.046	Shell Discard

TABLE 3.

TEST TO DETERMINE EFFECT OF HEATING TO A CHERRY RED ON THE STRENGTH OF THE REINFORCEMENT.

Portion of Bar Tested	Yield Point	Ultimate Strength	% Elongat'n in 8"	% Reduction in area
Unheated Portion	56,400	102,500	16.5	30.4
Junction of Heated & Unheated Portion	58,600	105,900	11.6	30.4
Heated Portion	56,400	102,500	15.3	34.6

Each figure is the average of two determinations.

Note:- The specifications call for a yield point of 50,000 and an ultimate of 80,000. It would appear from this table that heating to a cherry red did not change the tensile properties of the bars.

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TABLE 4

EFFECT OF FINENESS OF CEMENT ON THE STRENGTH OF CONCRETE

Each Figure is the average compressive Strength of twenty 4 x 8 Cylinders.

Atlas Aggregate Used: 1 volume Fine to 2 volumes Coarse; Graded up to 1/2":

Mix by volume; Same Brand of Cement used.

Relative consistencies from 1.00 to 1.50, the Gradation being the same in each case.

MIX	FINENESS (% CEMENT Passing a 200 sieve)	COMPRESSIVE STRENGTH 7 Days	COMPRESSIVE STRENGTH 28 days
1-4	91.2	1384	2200
	84.2	1170	1816
1-3	91.2	2040	3030
	84.2	1800	2538
1-2*	91.2*	3118*	4248*
	84.2	2642	3770
1-1-1/2	91.2	3540	4455
	84.2	3158	4378
1 - 1	91.2	4082	5090
	84.2	4014	4848

Tests by Prof. Duff Abrams, Lewis Institute, Chicago, Ill.

*Mix used for Concrete Ships

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To illustrate paper on "Methods Employed in the Construction of Concrete Ships," by R. J. Wig, Esq., Visitor.

TABLE NO. 5

COMPRESSIVE STRENGTH OF CONCRETE MADE FROM THREE TYPES OF LIGHTWEIGHT AGGREGATE MANUFACTURED FOR THE CONCRETE SHIP SECTION OF THE EMERGENCY FLEET CORPORATION.

MANUFACTURERS OF AGGREGATE AND TYPE OF KILN USED IN BURNING.	BRAND OF CEMENT	MIX	CONSISTENCY DROP IN INCHES.	WT. OF CON-CRETE LBS. per. cu. ft.	AGE WEHN TESTED IN DAYS	COMPRESSIVE STRENGTH OF CYLINDERS 6" DIAM. X 12" HIGH.	MODULUS OF ELASTICITY.
Atlas Portland Cement Company, Hannibal, Mo. Burned in Cement Kiln	Southern States Reground so 90% passes a #200 sieve.	1 Cement 1 agg. under 1/10". 1 agg. between 1/10" and 1/2".	9.12	119.3	28	4537	2,953,000
Los Angeles Pressed Brick Company., Burned in Brick Kiln (See Fig. ~~80~~ 40).	Santa Cruz Reground.	1 Cement 2/3 agg. under 1/10" - 1-1/3 agg. between 1/10 and 1/2"	9.19	105.2	28	3861	----
		1 Cement 2/3 agg. under 1/10" - 1-1/3 agg. between 1/10" & 1/2" Celite *	9.91	102.1	28	3256	----
Copland-Inglis Co., Birmingham, Ala Burned in brick kiln.	Giant Reground	1 cement 2/3 agg. under 1/10 - 1-1/3 agg. between 1/10" & 1/2"	8.50	115.6	33	3576	2,800,000

* Diatomaceous Earth :- 1-1/2% by weight of Cement.

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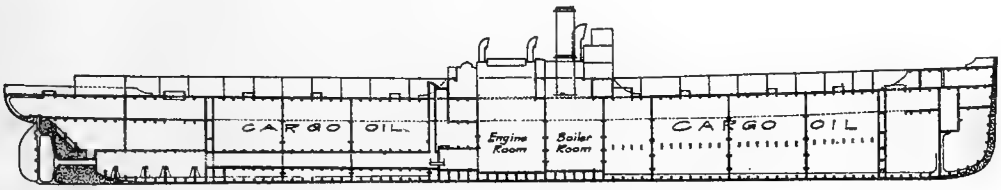
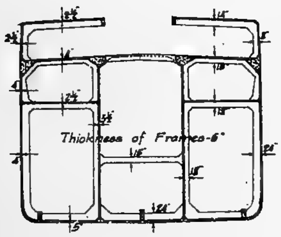


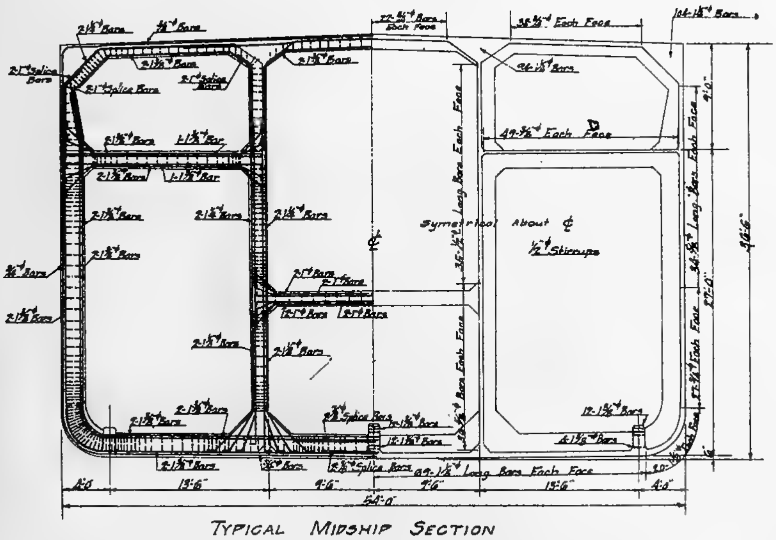
INBOARD PROFILE



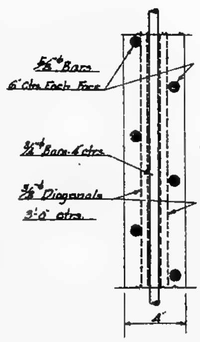
MIDSHIP SECTION

LENGTH OVER ALL	434'- 3"
LENGTH BETWEEN PERPENDICULARS	420'- 0"
BREADTH OVER ALL	54'- 0"
DEPTH MOULDED AT SIDE TO UPPER DK.	36'- 0"
DESIGNED LINE OF LOAD DRAFT (FULL)	26'- 0"
CAMBER OF DECKS	12"
FRAME SPACING	4'- 3"

7500 TON CONCRETE TANKER
INBOARD PROFILE
AND
MIDSHIP SECTION
DESIGN No. 70



TYPICAL MIDSHIP SECTION
Showing Frame Steel



SECTION OF SIDE SHELL
SHOWING REINFORCEMENT

Note:- Section Taken at Right Angles to Base Line

7500 TON CONCRETE TANKER
TYPICAL MIDSHIP SECTION
AND
SECTION OF SHELL
SHOWING REINFORCING
DESIGN No. 70

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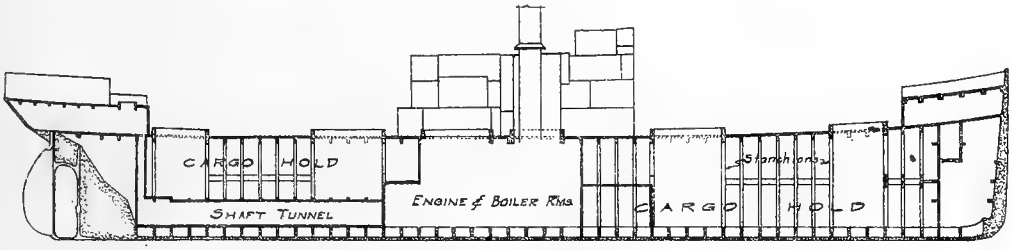
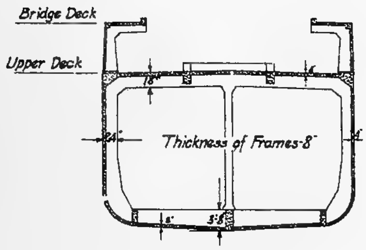


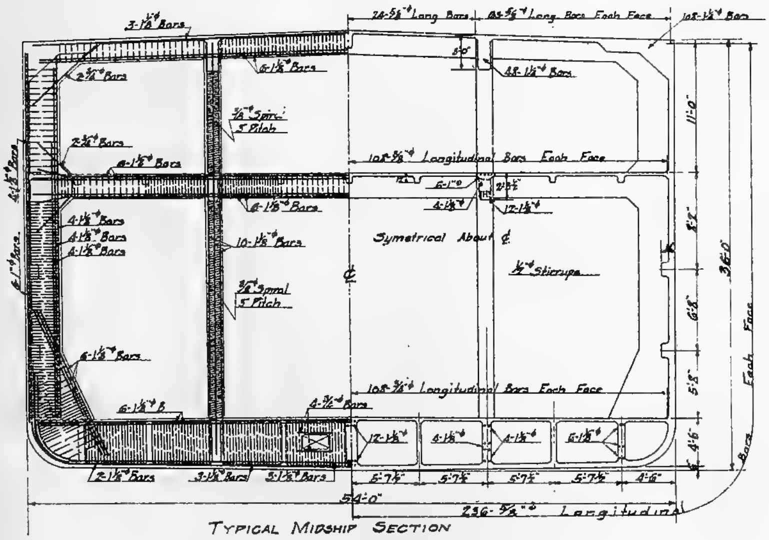
INBOARD PROFILE



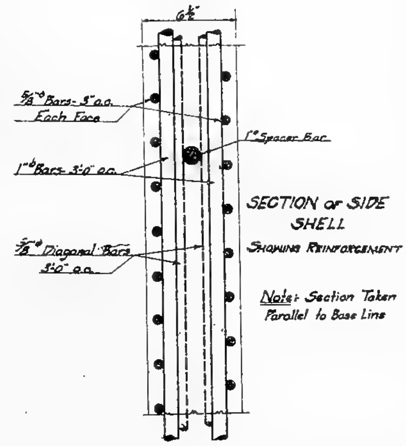
MIDSHIP SECTION

LENGTH OVER ALL ——— 281'- 10"
LENGTH BETWEEN PERPS ——— 268'- 0"
BREADTH OVER ALL ——— 46'- 0"
DEPTH MOULDED AT SIDE TO UPPER DK. — 28'- 3"
DESIGNED LINE OF LOAD DRAFT (FULL) — 24'- 0"
CAMBER OF DECKS - 9" IN 46'-0"
FRAME SPACING ——— 5'- 0"

3500 TON CONCRETE CARGO SHIP
INBOARD PROFILE
AND
MIDSHIP SECTION
DESIGN No. 2



TYPICAL MIDSHIP SECTION
Showing Frame Steel



SECTION OF SIDE SHELL
SHOWING REINFORCEMENT

Note: Section Taken Parallel to Base Line

7500 TON CONCRETE CARGO SHIP
TYPICAL MIDSHIP SECTION
AND
SECTION OF SHELL
SHOWING REINFORCING
DESIGN No. 69

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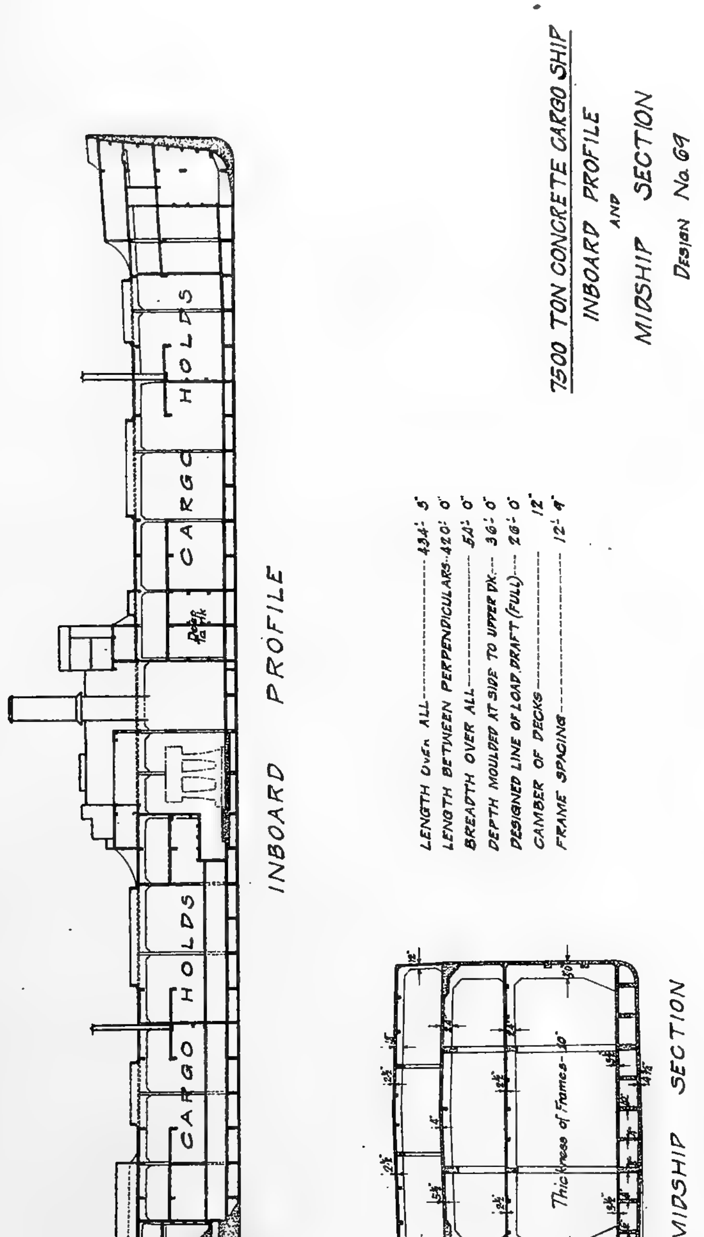

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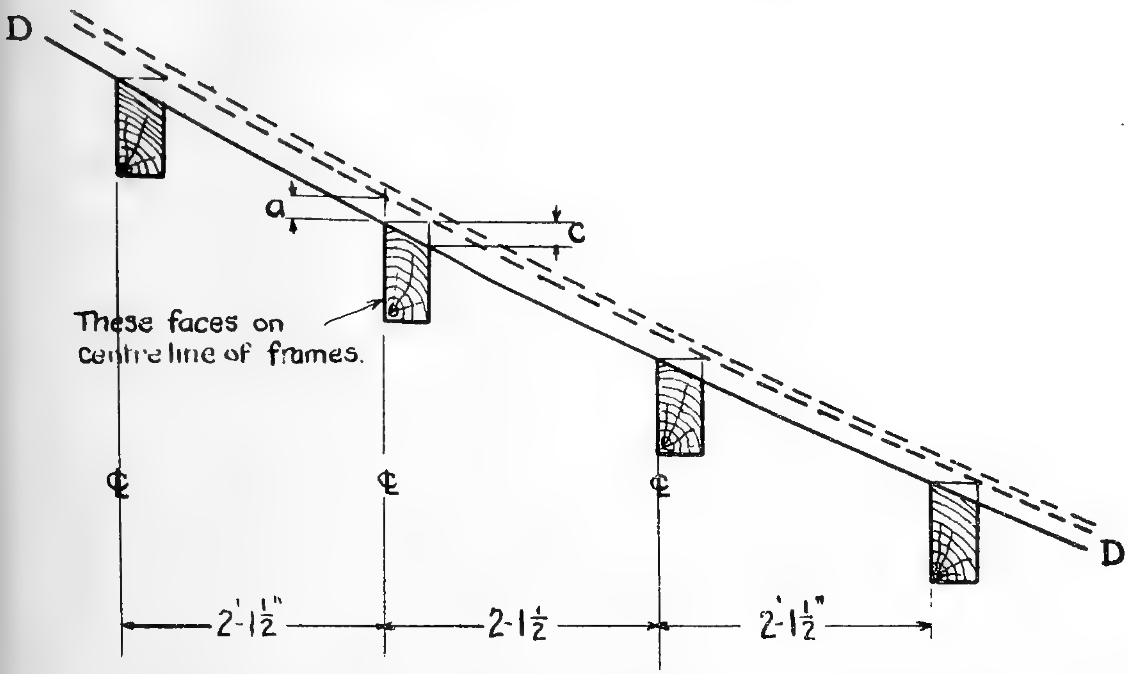
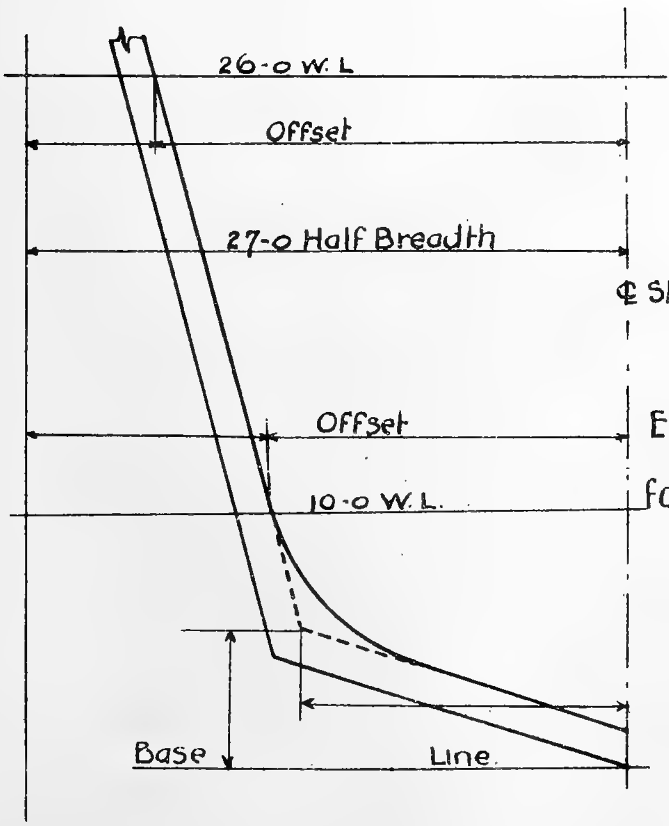
To illustrate paper on "Methods Employed in the Construction of Concrete Ships," by R. J. Wig, Esq., Visitor.



Plan of 2×6 Verticals for Outside Forms.



Elevation of 2×6 Verticals for outside forms

FIG 28

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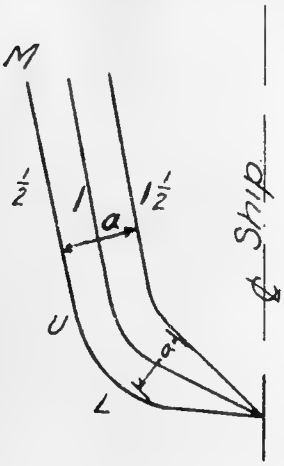

FIG 30 a

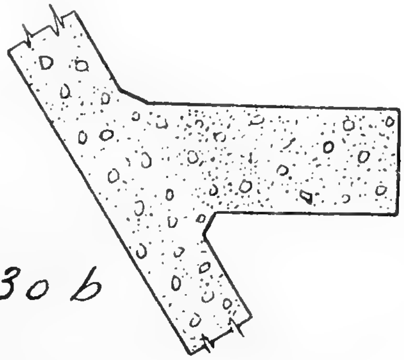

FIG 30 b

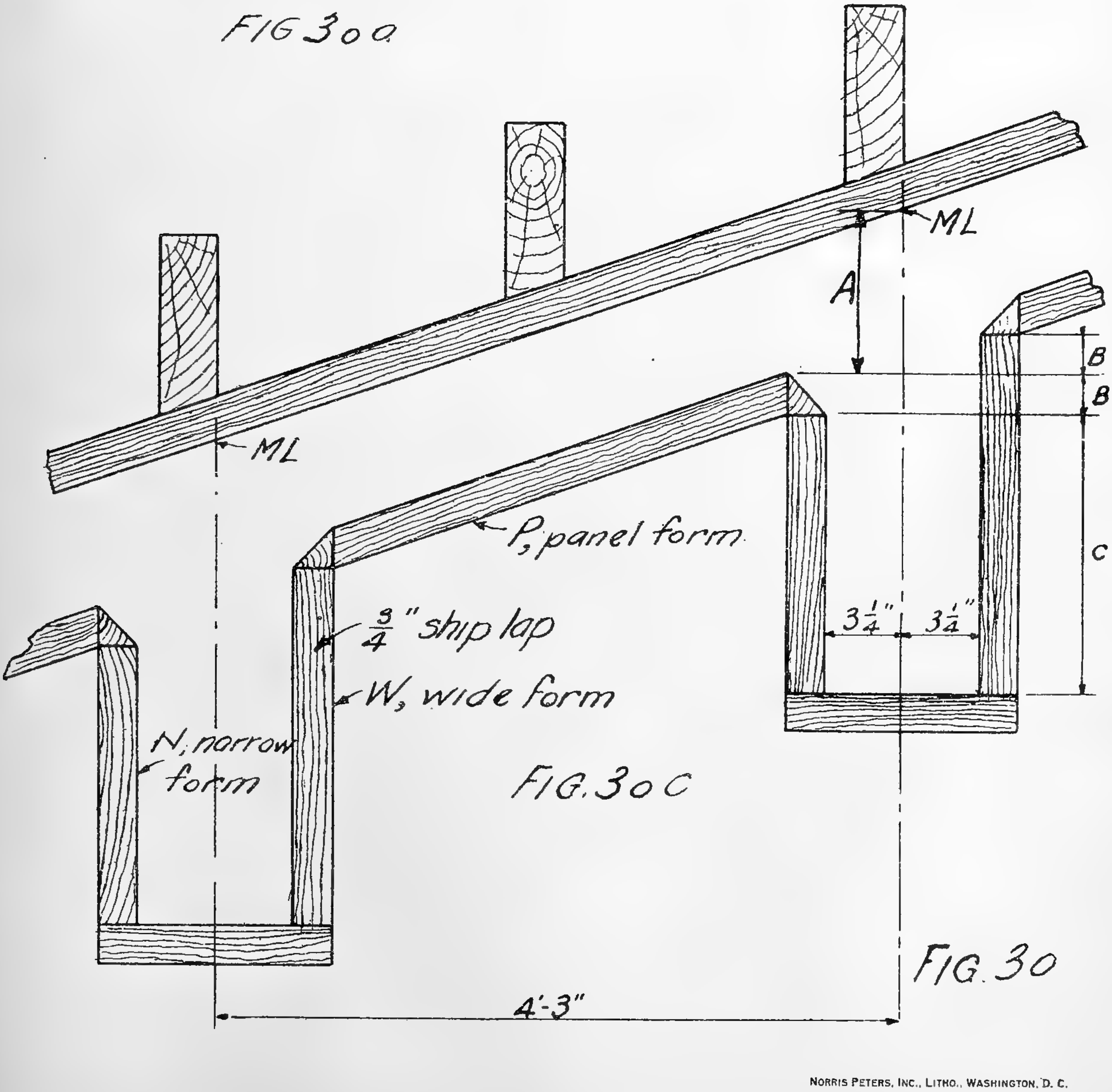

FIG. 30 C

FIG. 30

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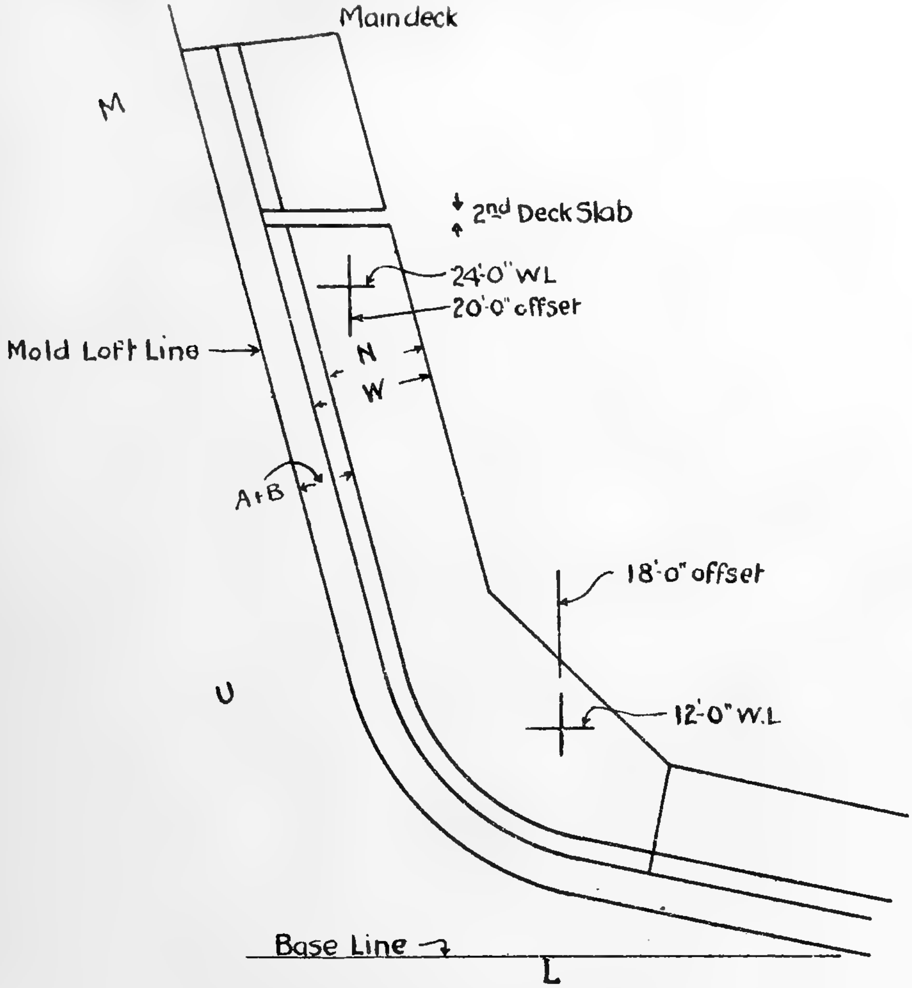
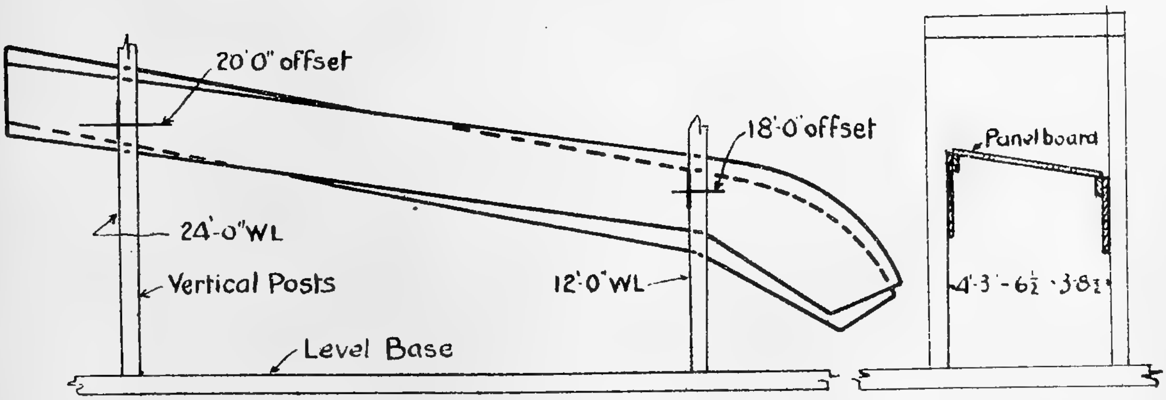


PLAN OF FRAME FORMS IN POSITION ON MOLD LOFT FLOOR.



FRAME FORMS IN POSITION IN RIG FOR ASSEMBLING AND BRACING FORMS. *FIG 31*

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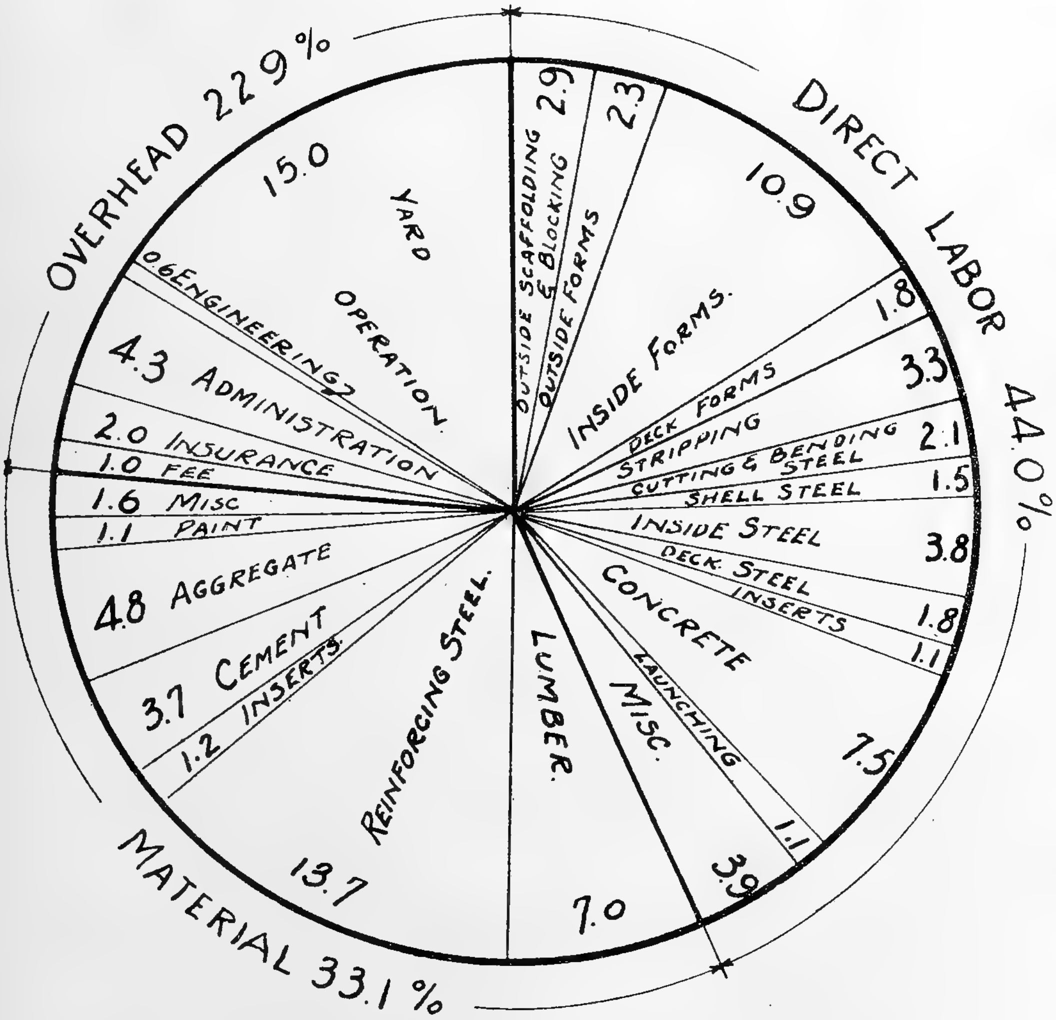
To illustrate paper on "Methods Employed in the Construction of Concrete Ships," by R. J. Wig, Esq., Visitor.

FIG 43



DISTRIBUTION OF COSTS HULL 1715
TOTAL COST OF HULL $786,754.

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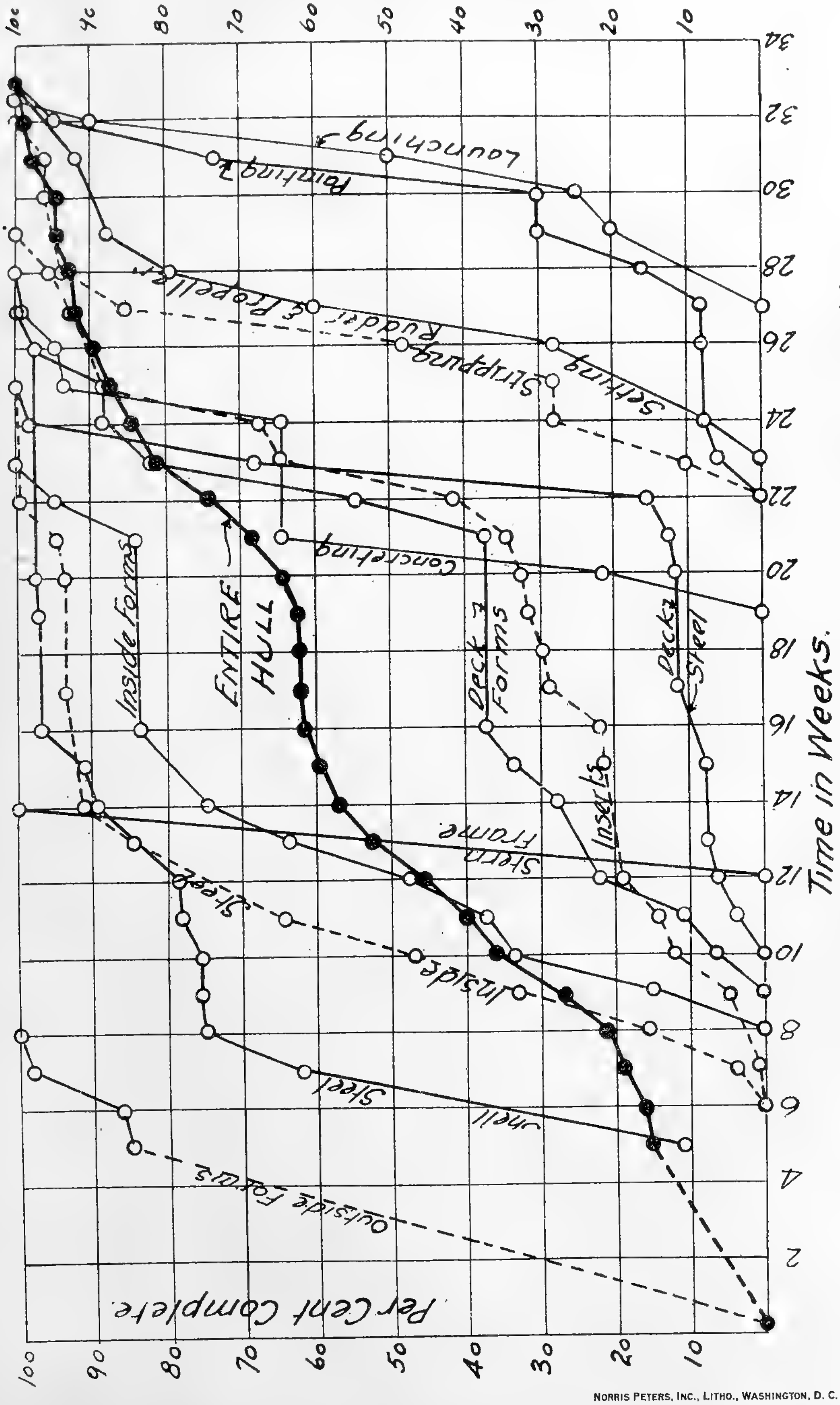
To illustrate paper on "Methods Employed in the Construction of Concrete Ships," by R. J. Wig, Esq., Visitor.

FIG. 44.



PROGRESS OF CONSTRUCTION ON HULL 1663.

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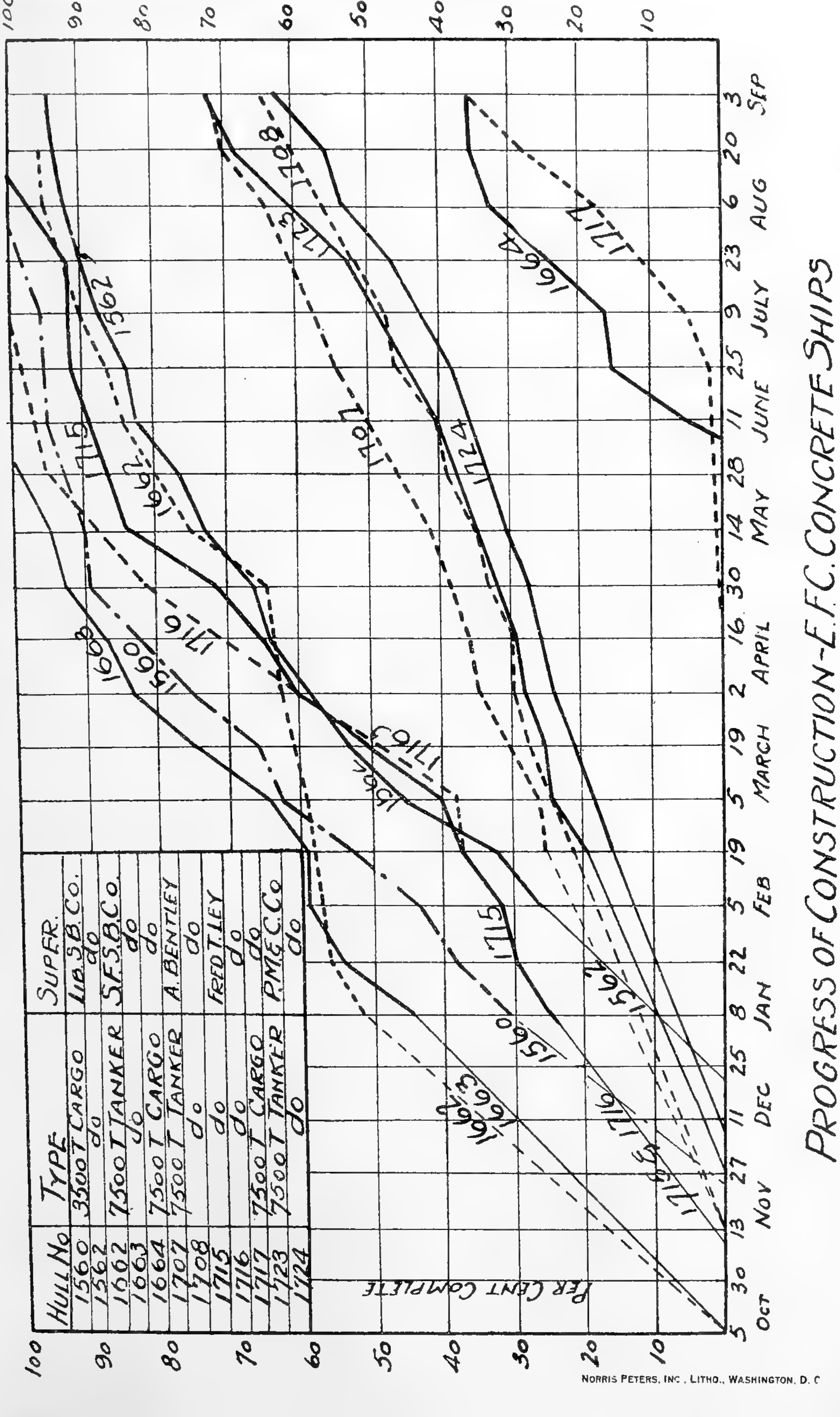
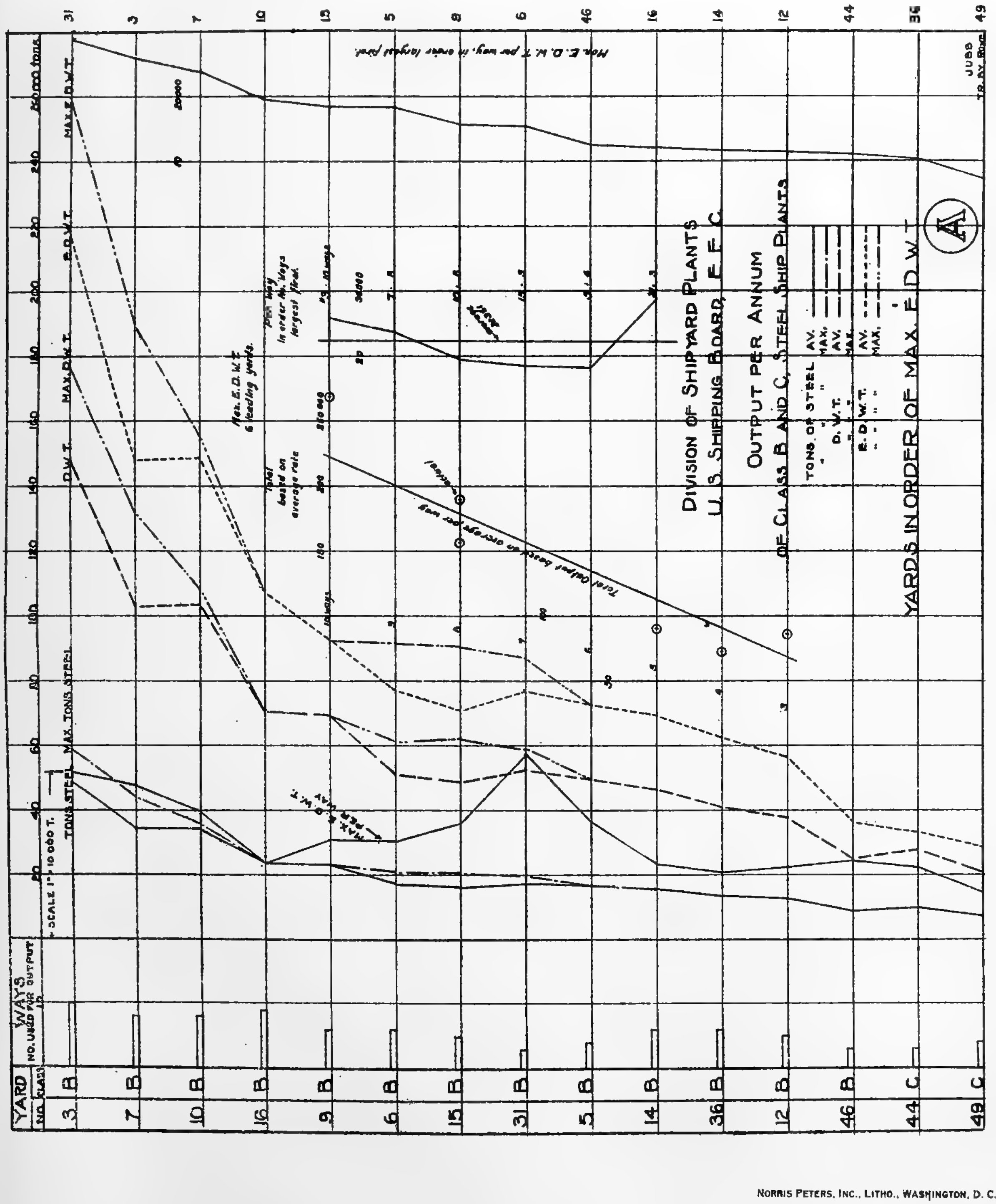
FIG. 45.



To illustrate paper on "Development of Shipyards in the United States During the Great War," by Captain R. E. Bakenhus, C. E. C., U. S. N., Visitor.



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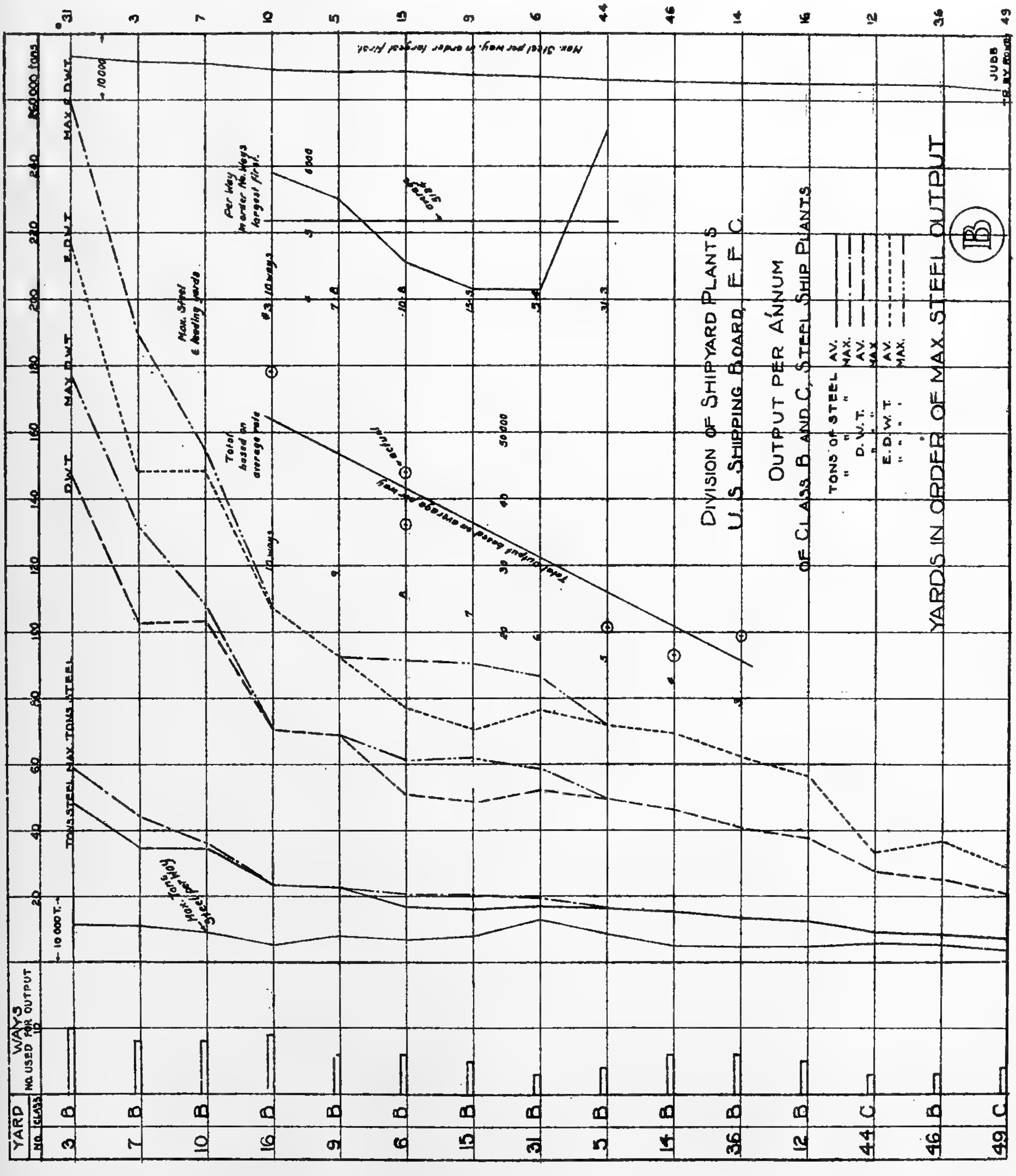
To illustrate paper on "Development of Shipyards in the United States During the Great War," by Captain R. E. Bakenhus, C. E. C., U. S. N., Visitor.



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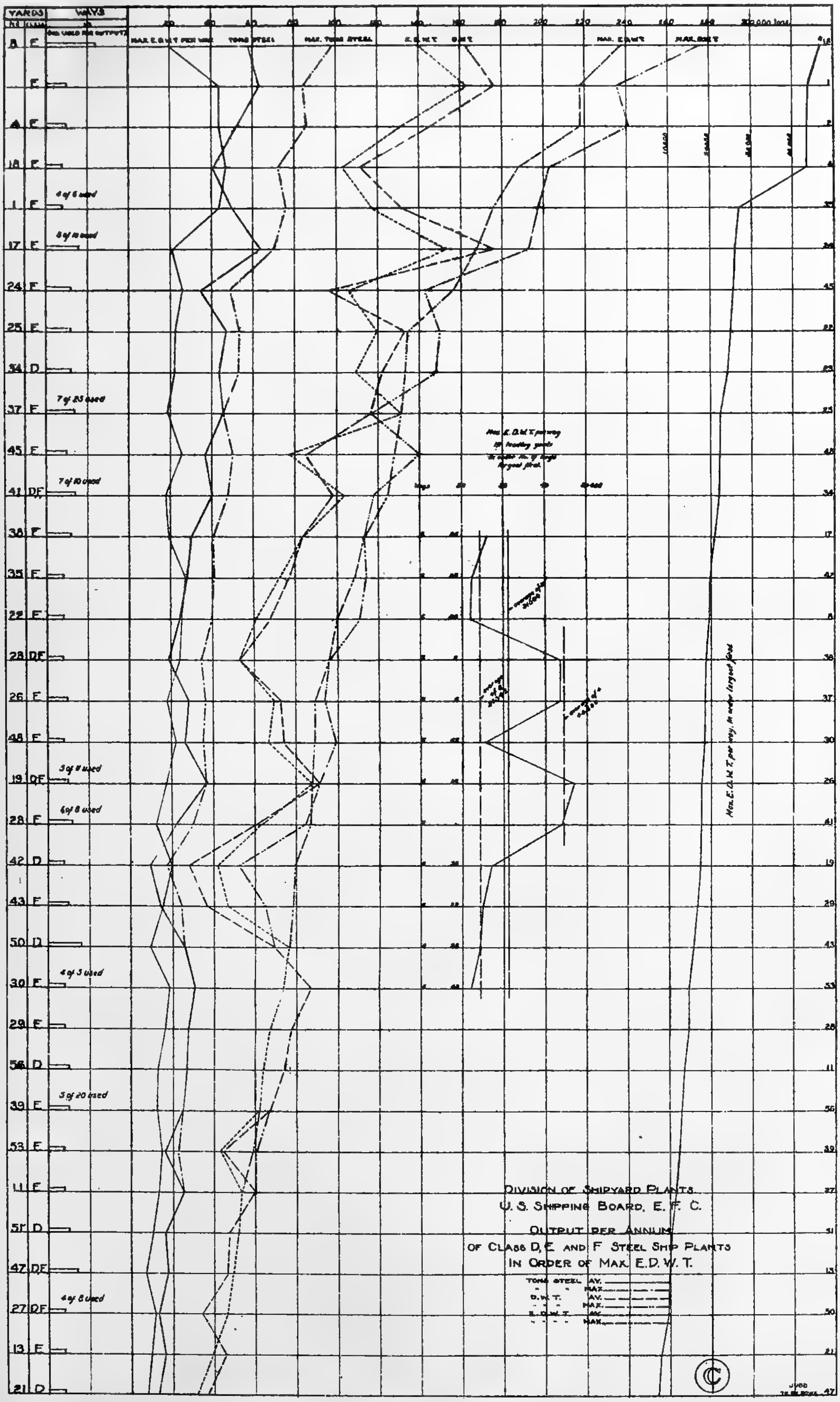

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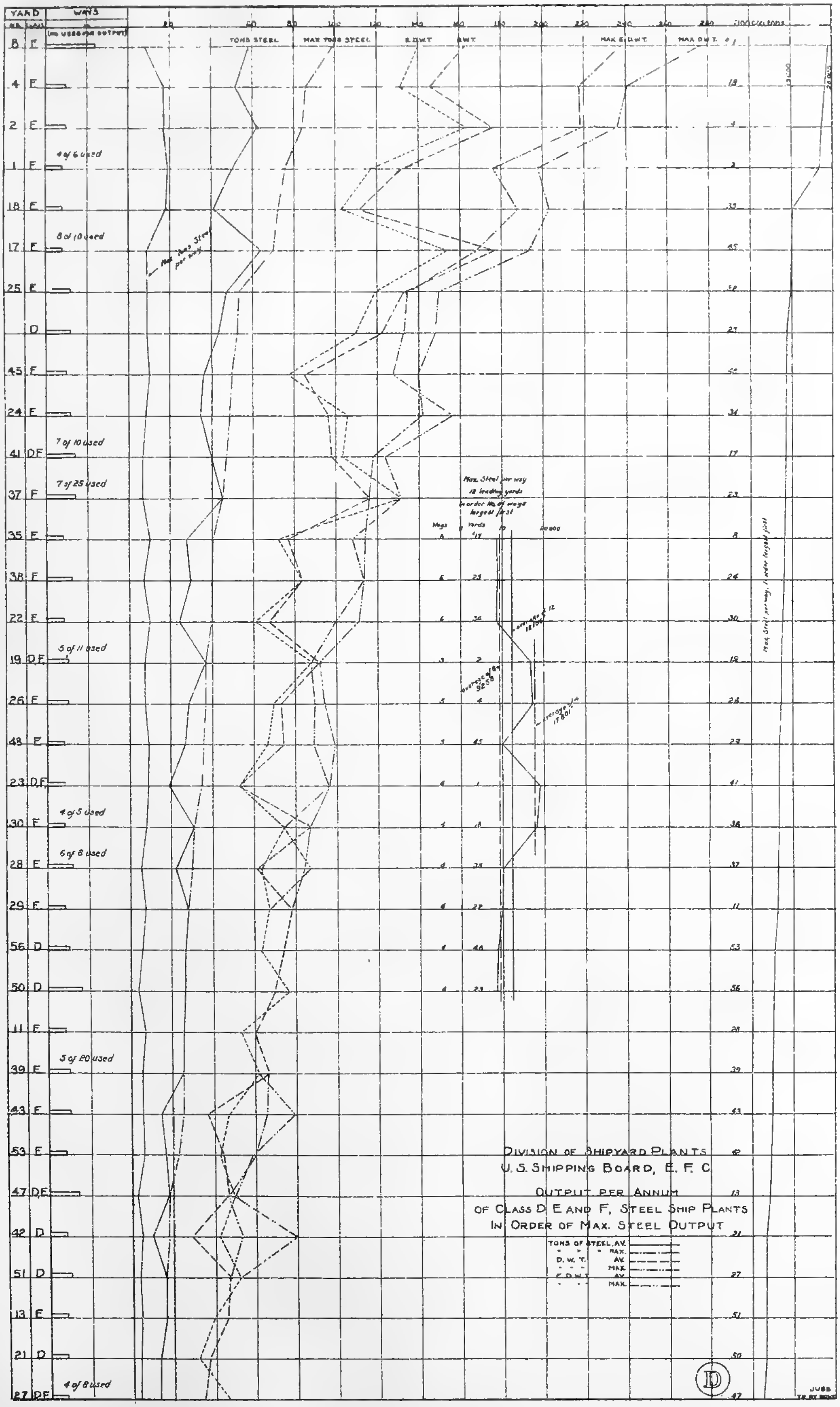

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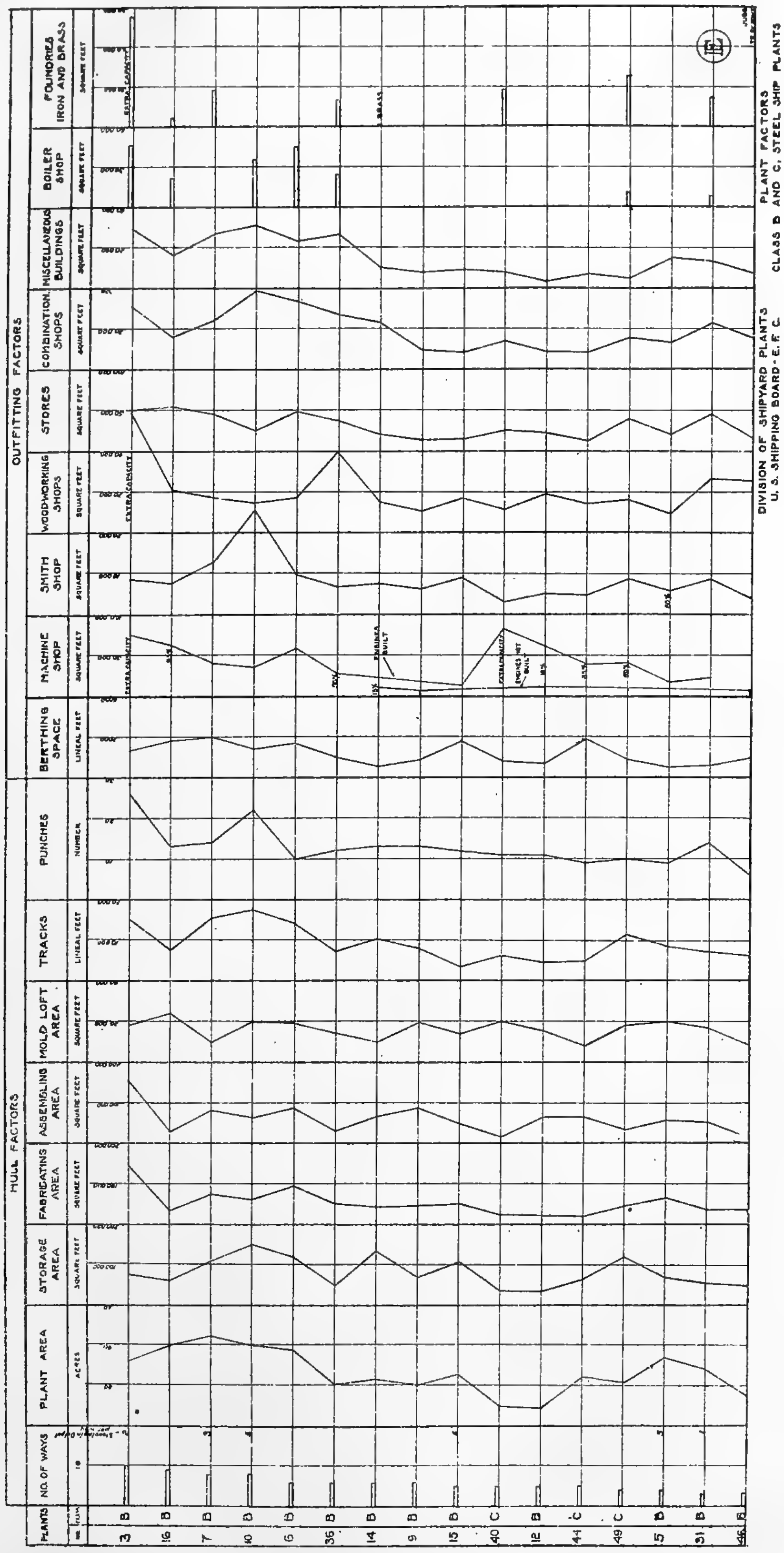

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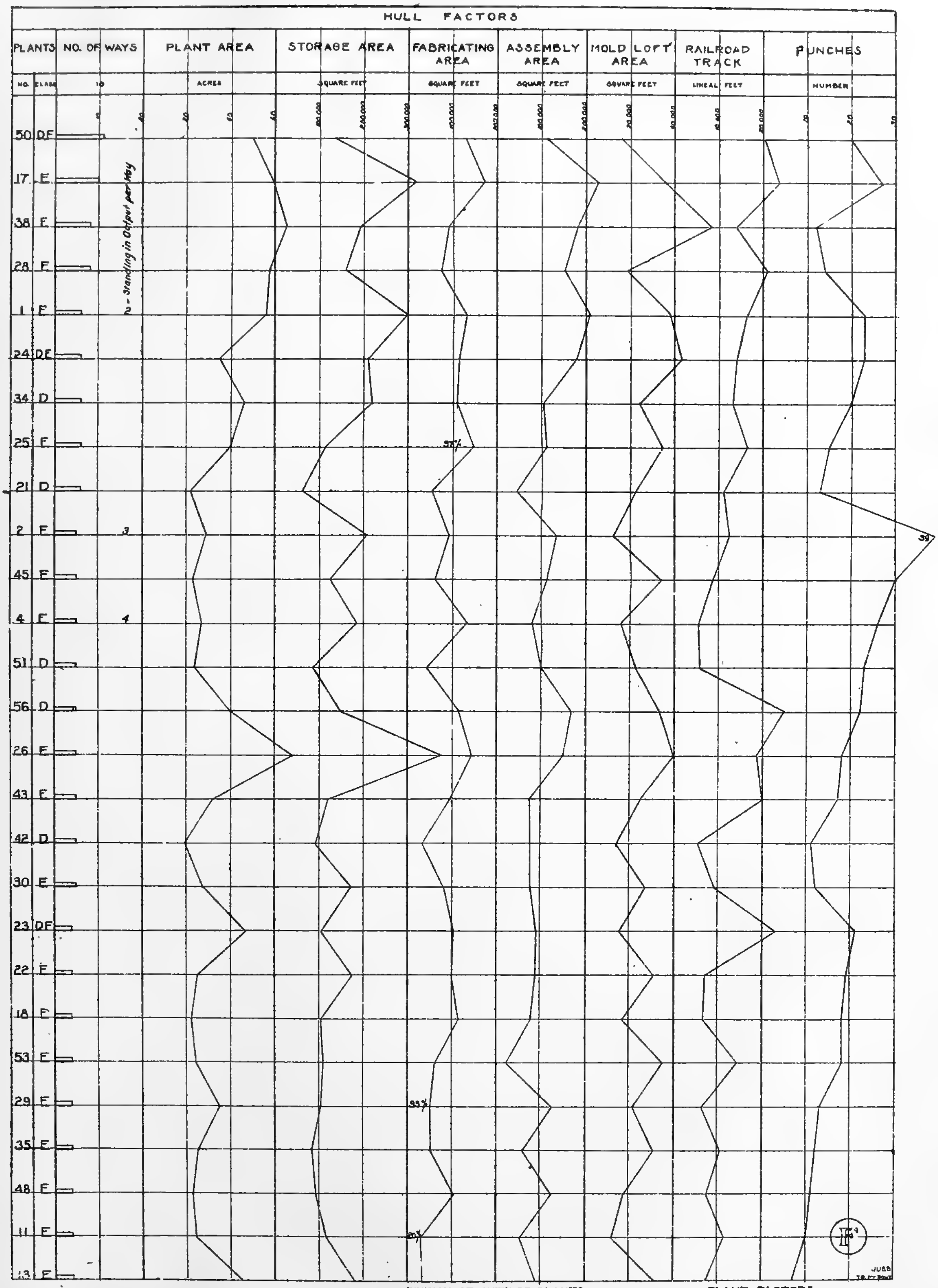

DIVISION OF SHIPYARD PLANTS
U. S. SHIPPING BOARD—E. F. C.

PLANT FACTORS
CLASS D. E AND F. STEEL SHIP PLANTS

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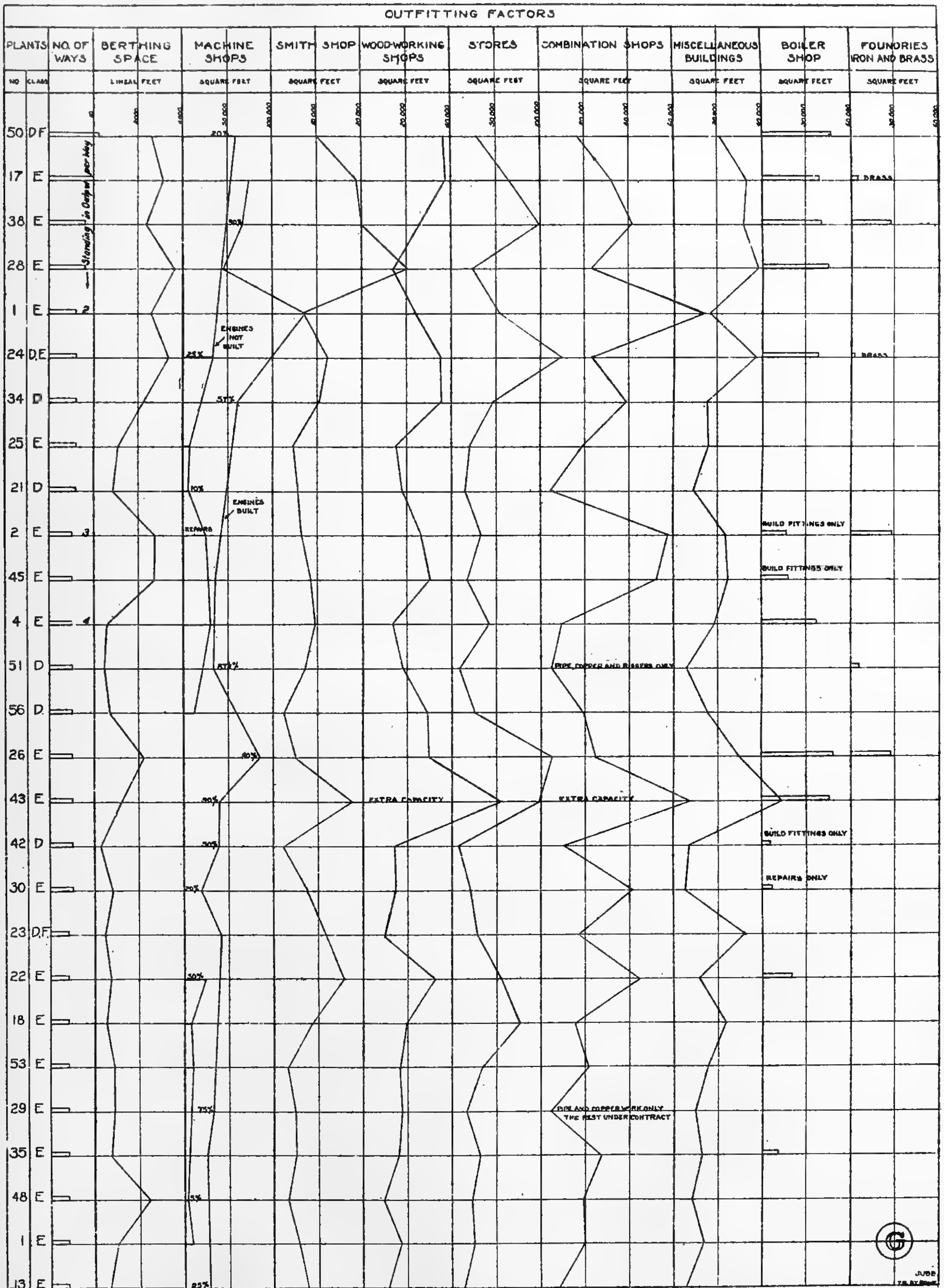
To illustrate paper on "Development of Shipyards in the United States During the Great War," by Captain R. E. Bakenhus, C. E. C., U. S. N., Visitor.



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PLANT FACTORS
CLASS D, E AND F, STEEL SHIP PLANTS

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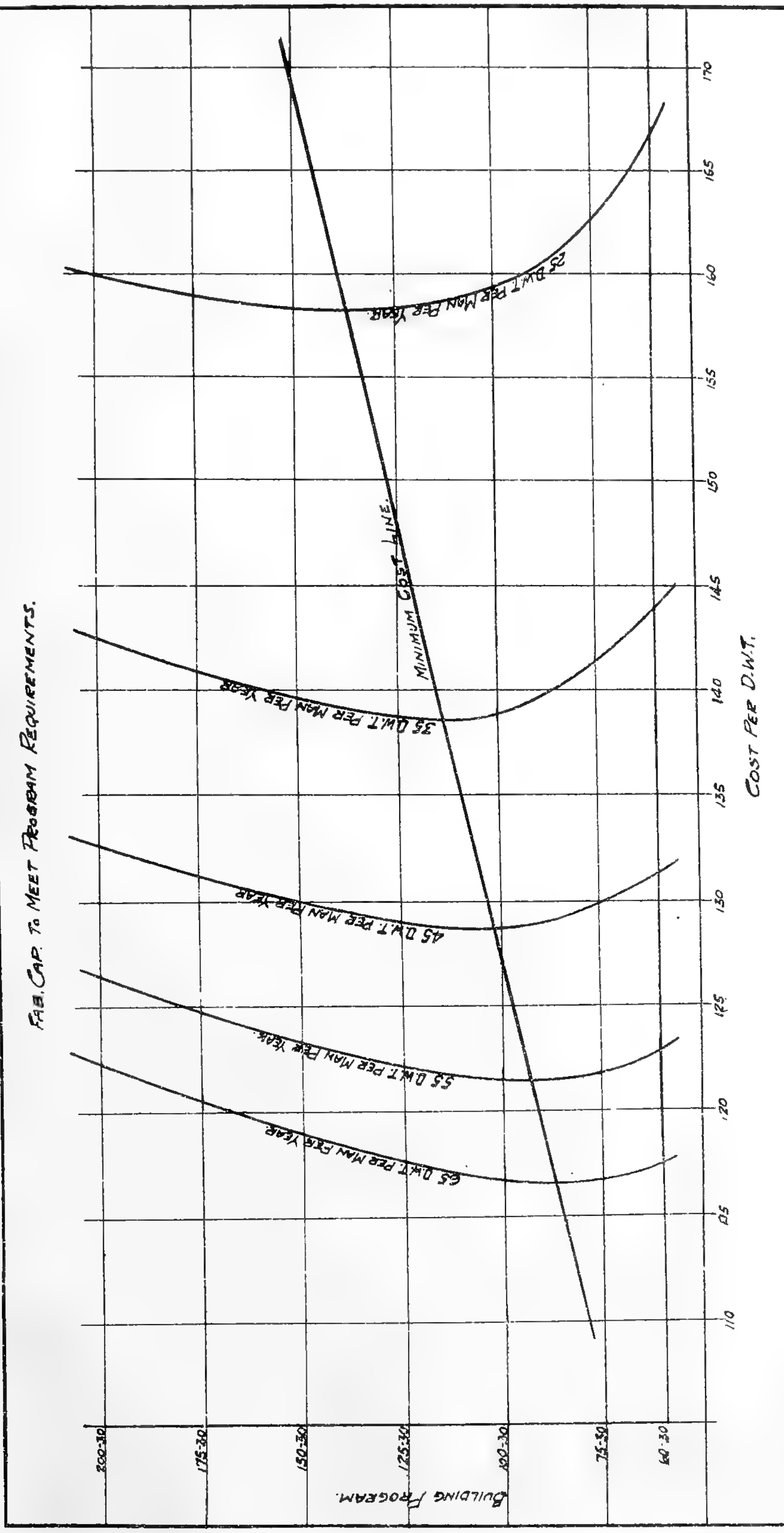
To illustrate paper on "Steel Ship Construction from a Management View Point," by Creighton Churchill, Esq., Member.



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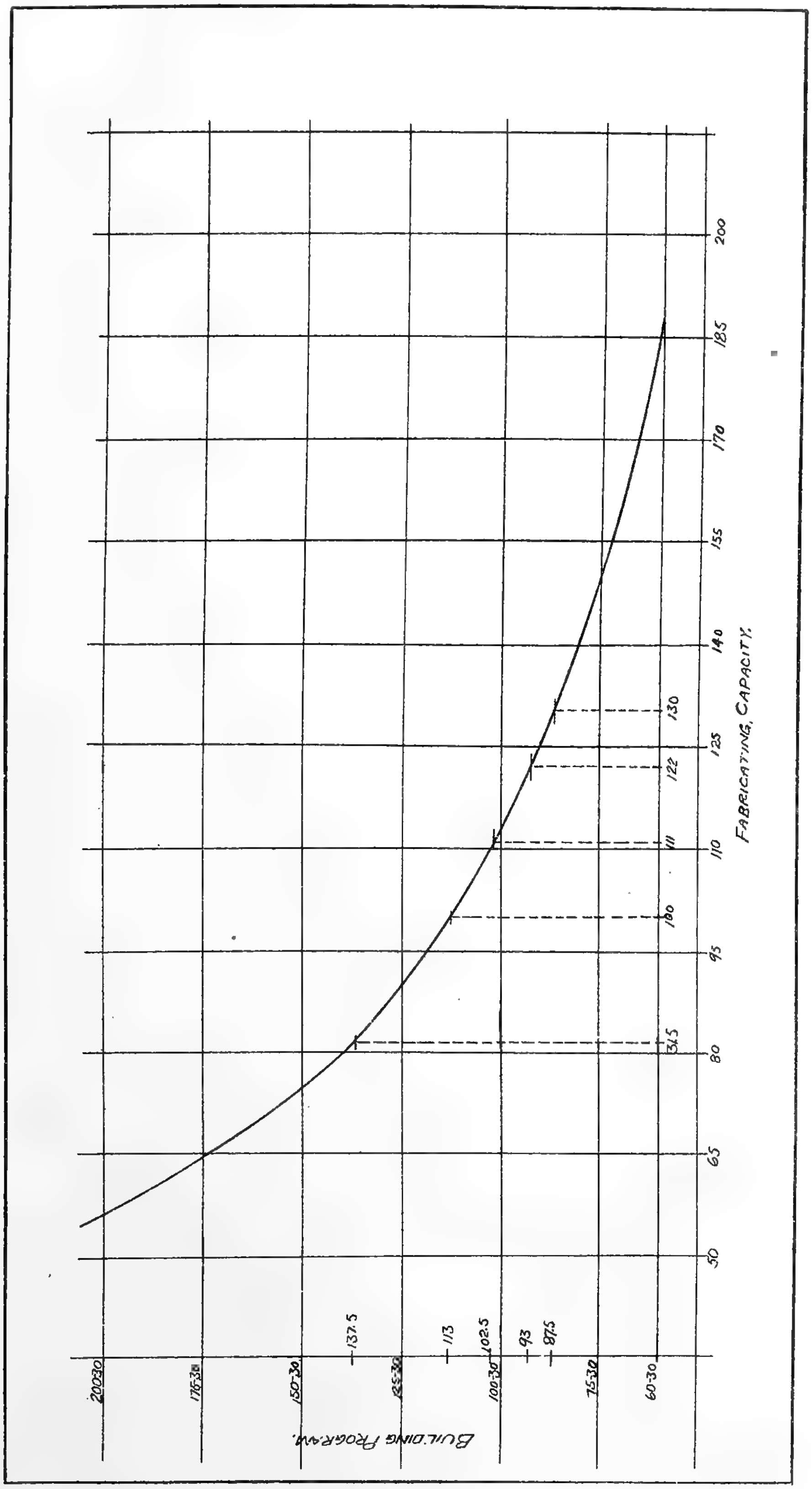

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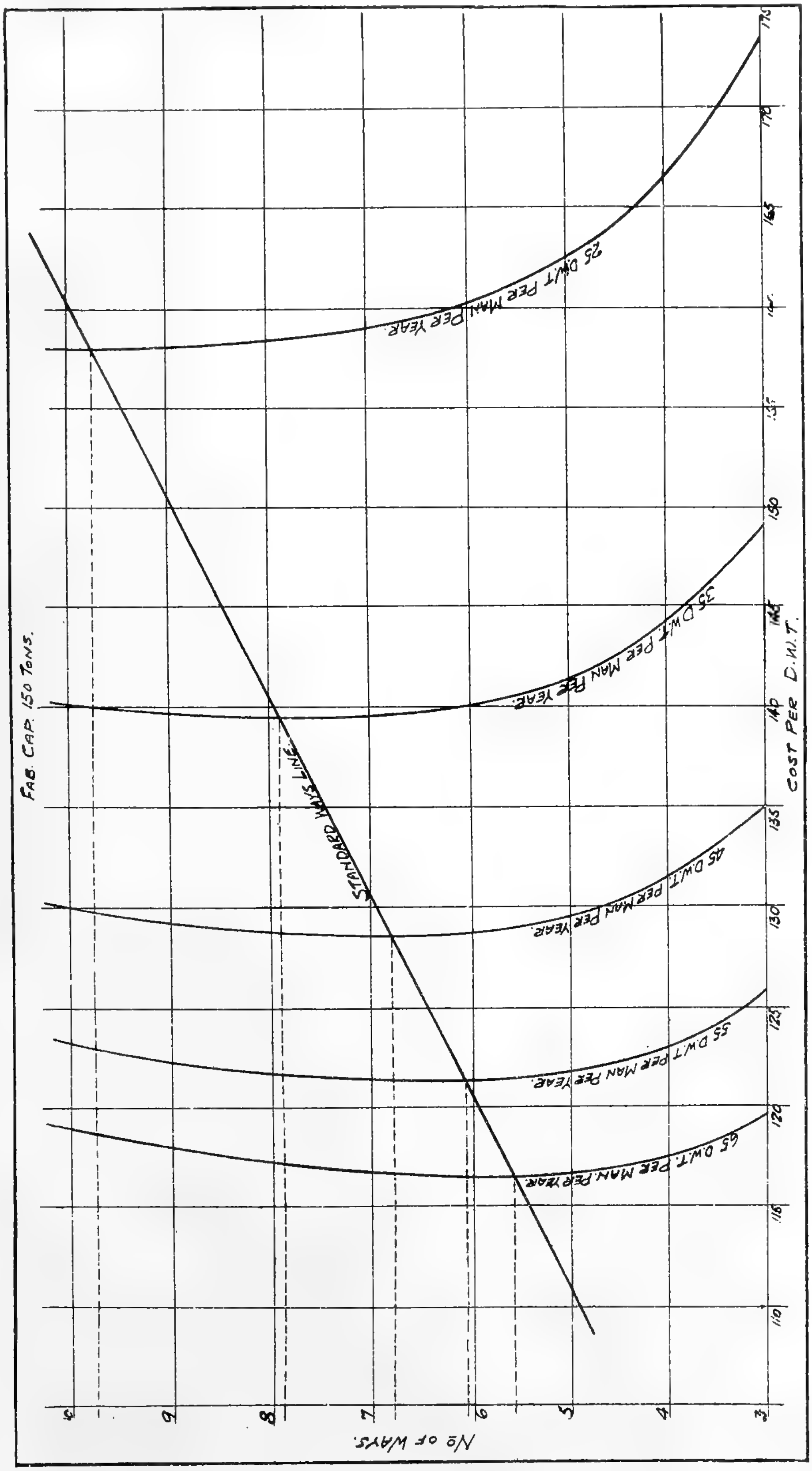

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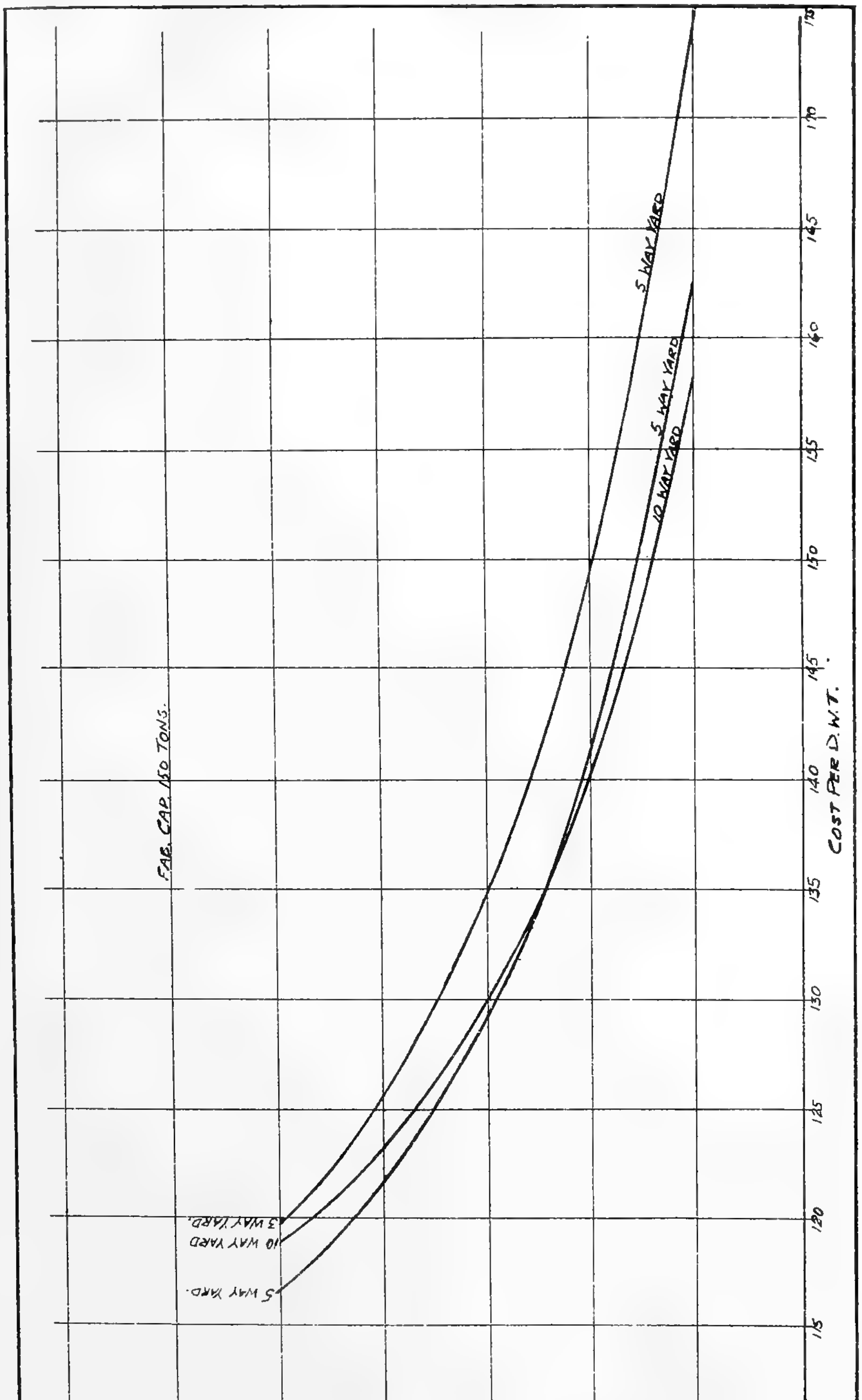

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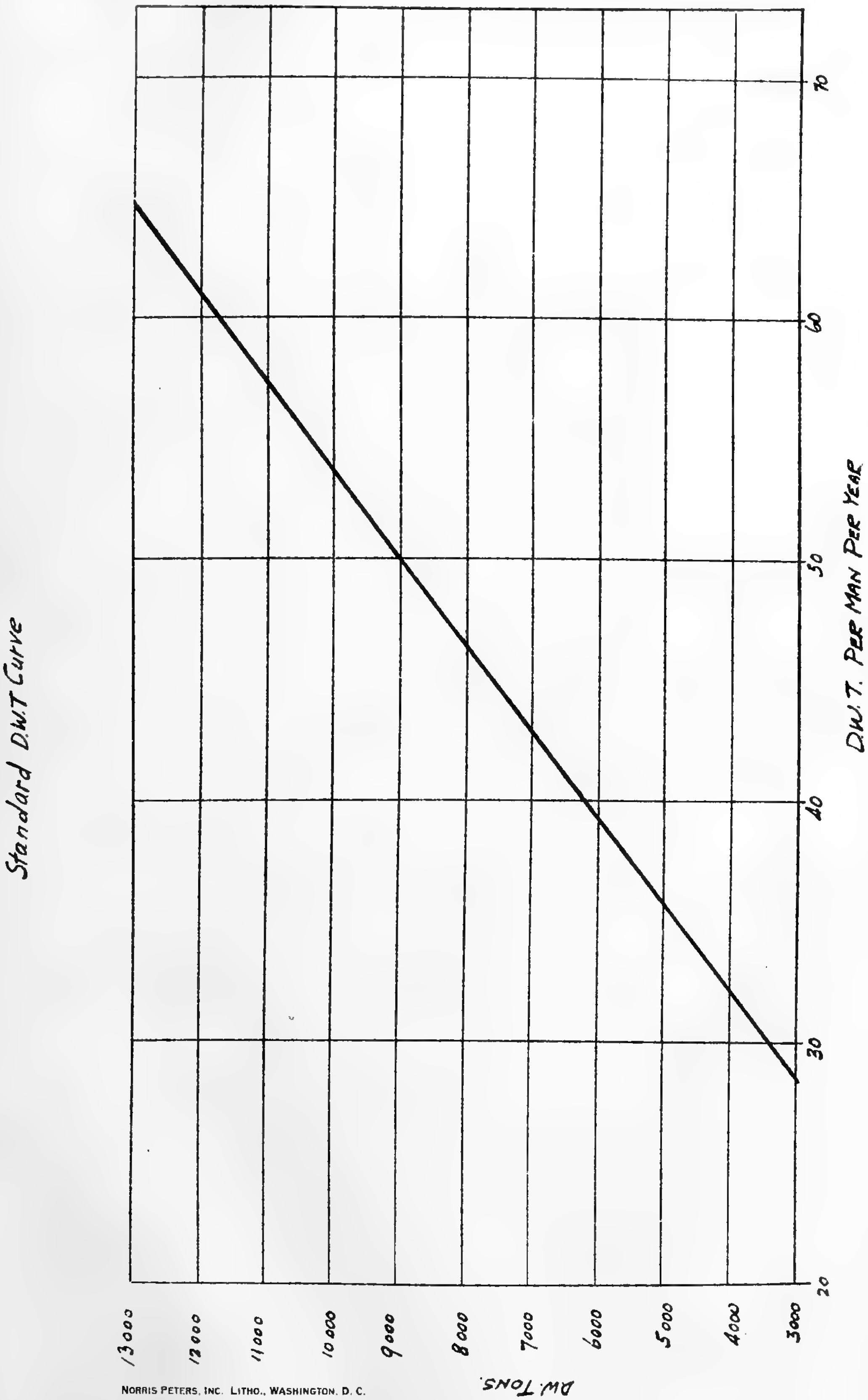

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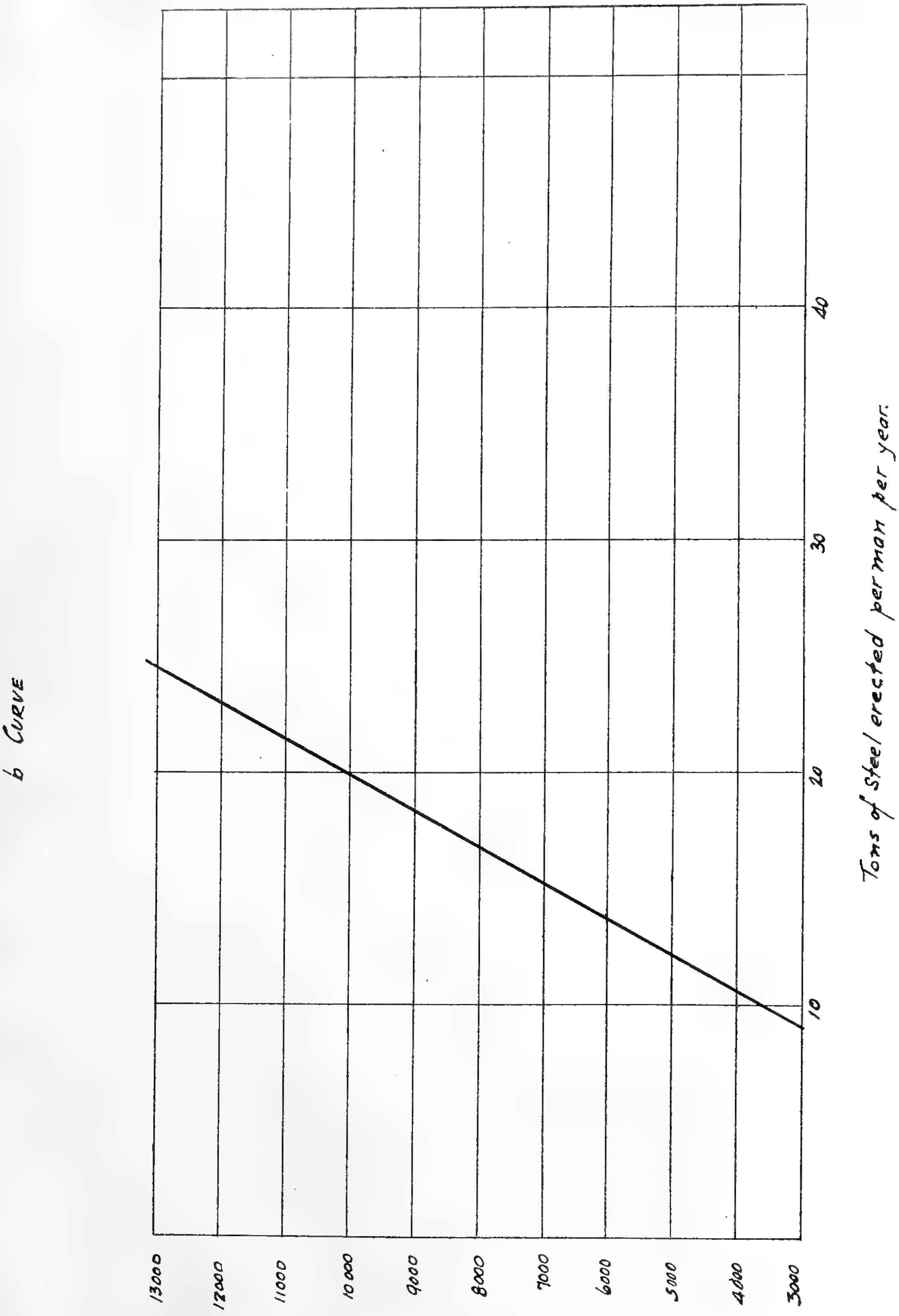

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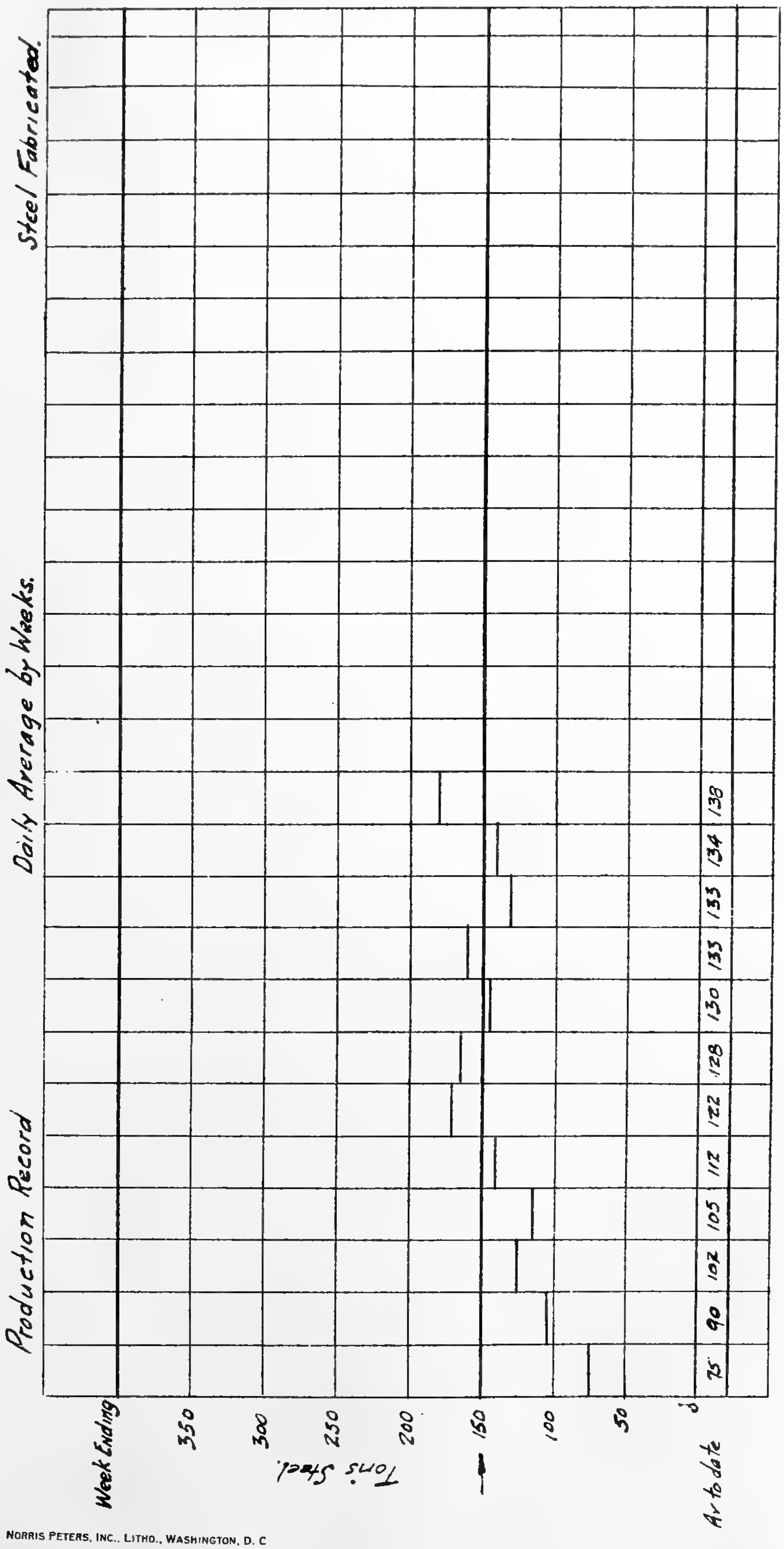

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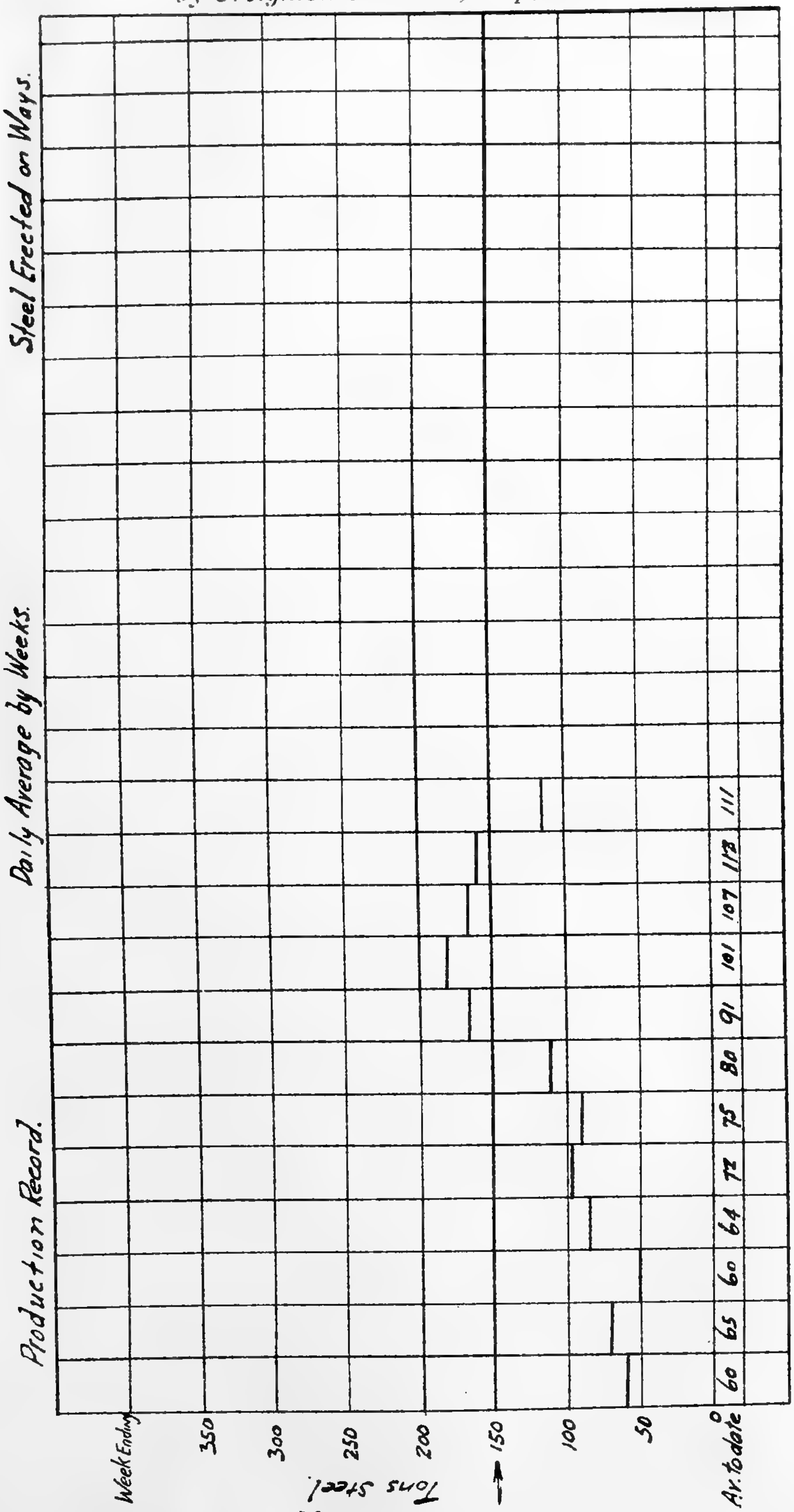

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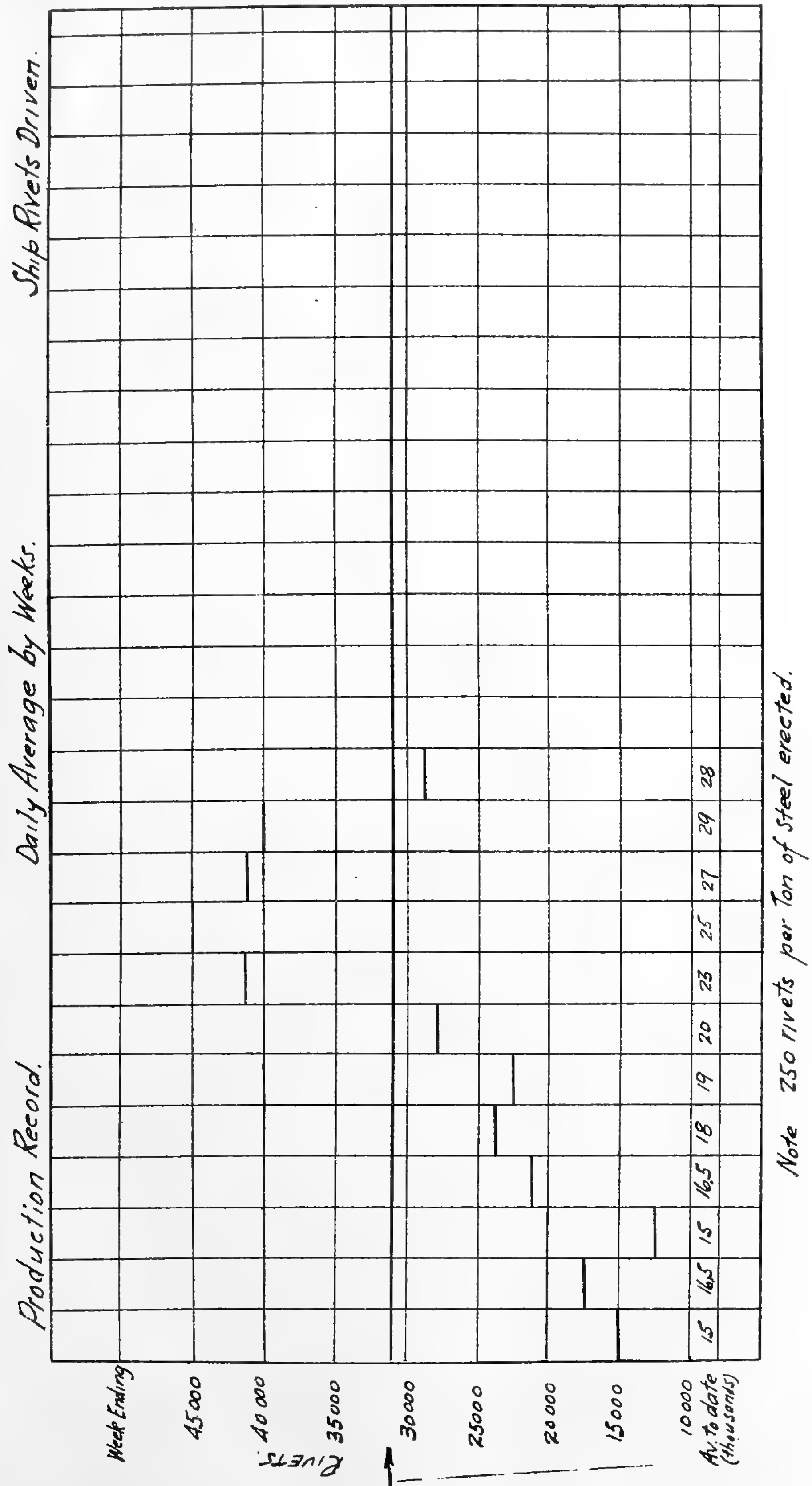

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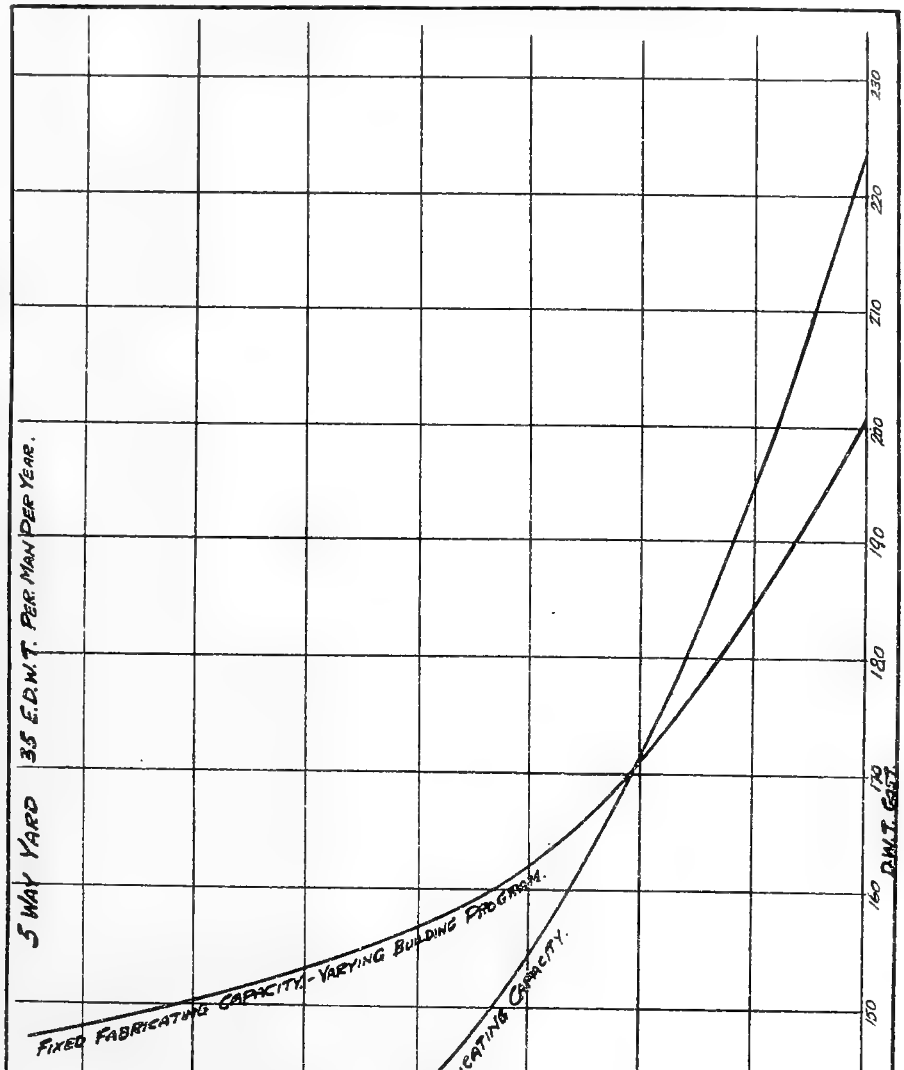

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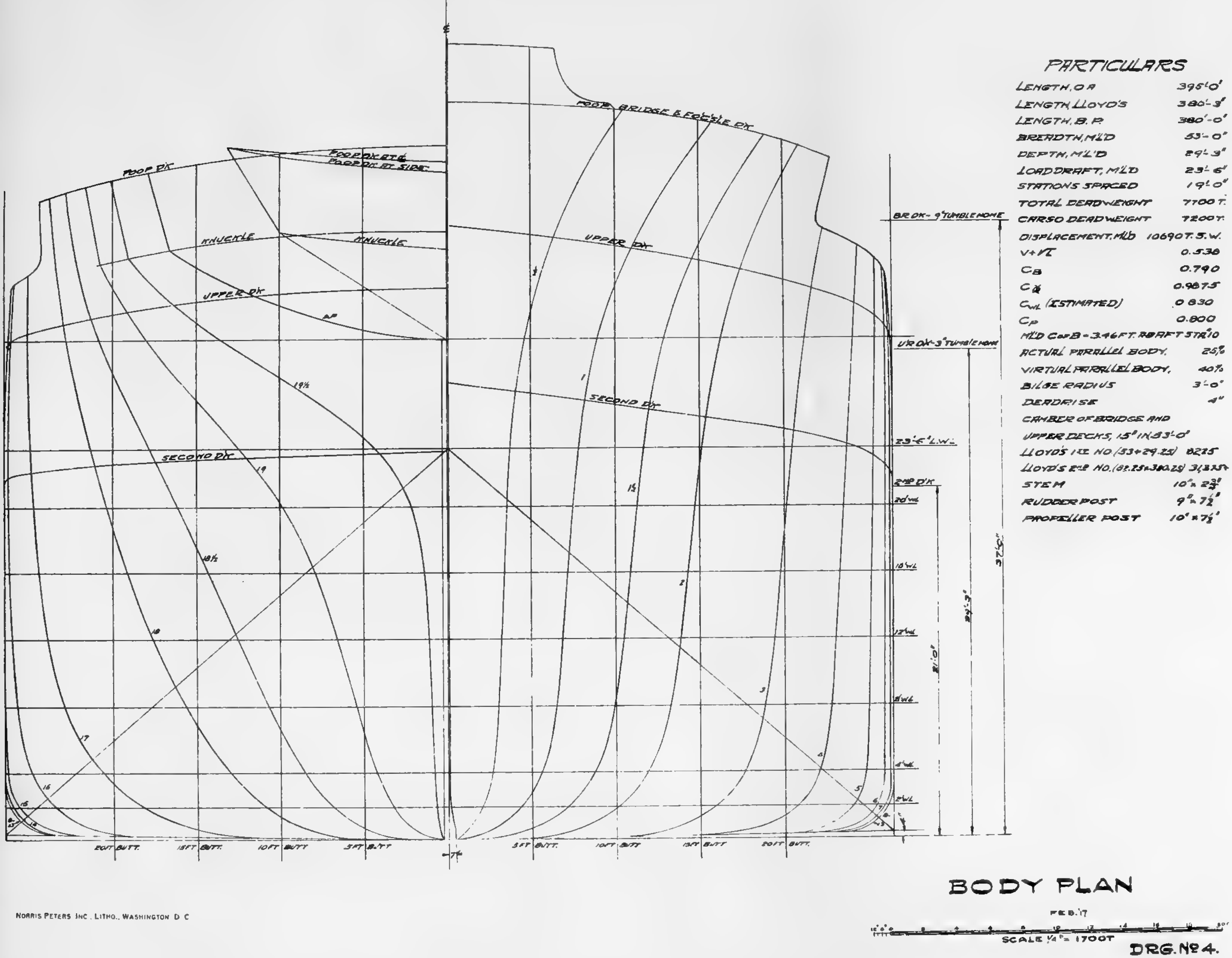
To illustrate paper on "An Analysis of the Isherwood System of Ship Construction," by John Flodin, B. S., M. E., Associate.



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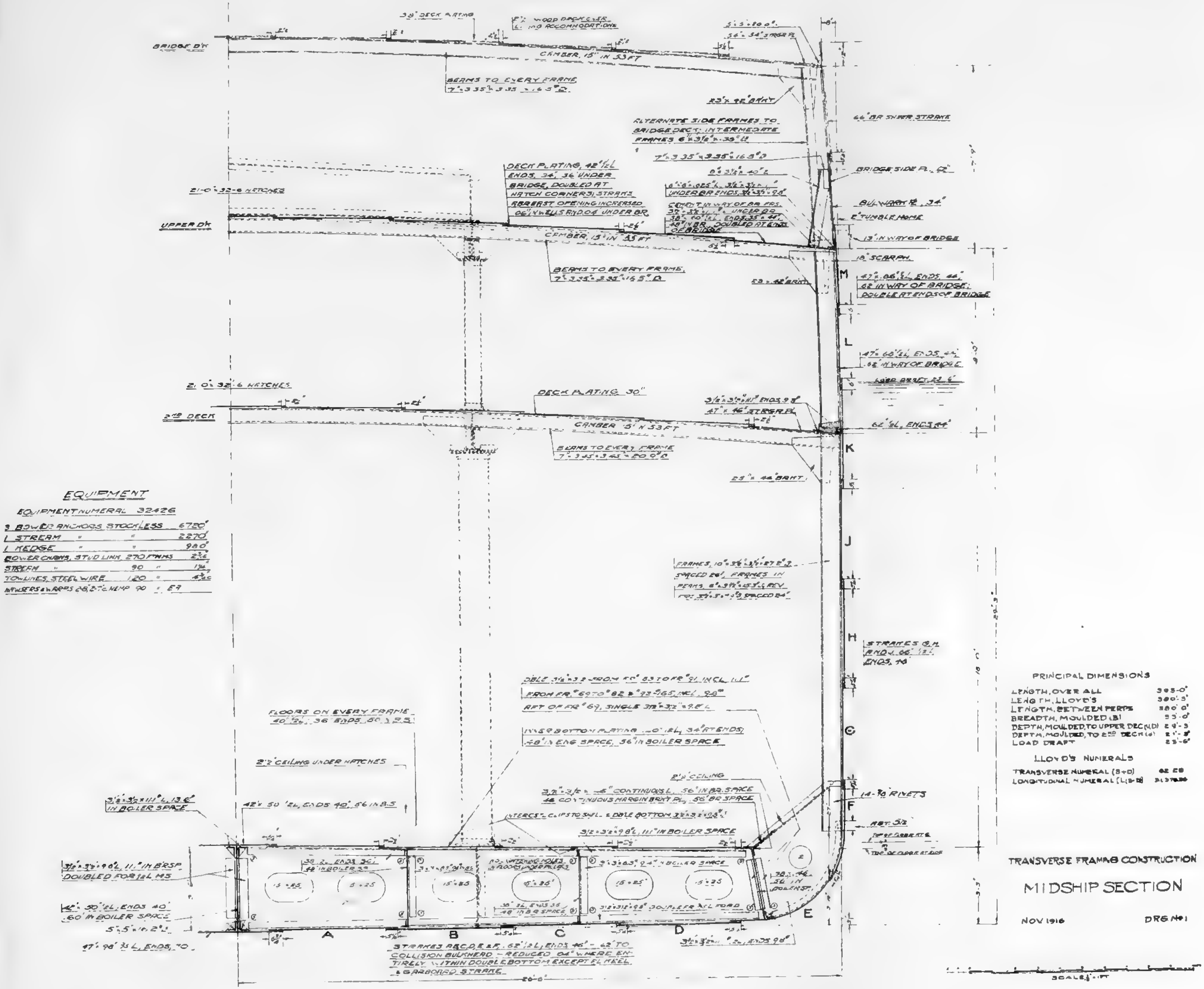

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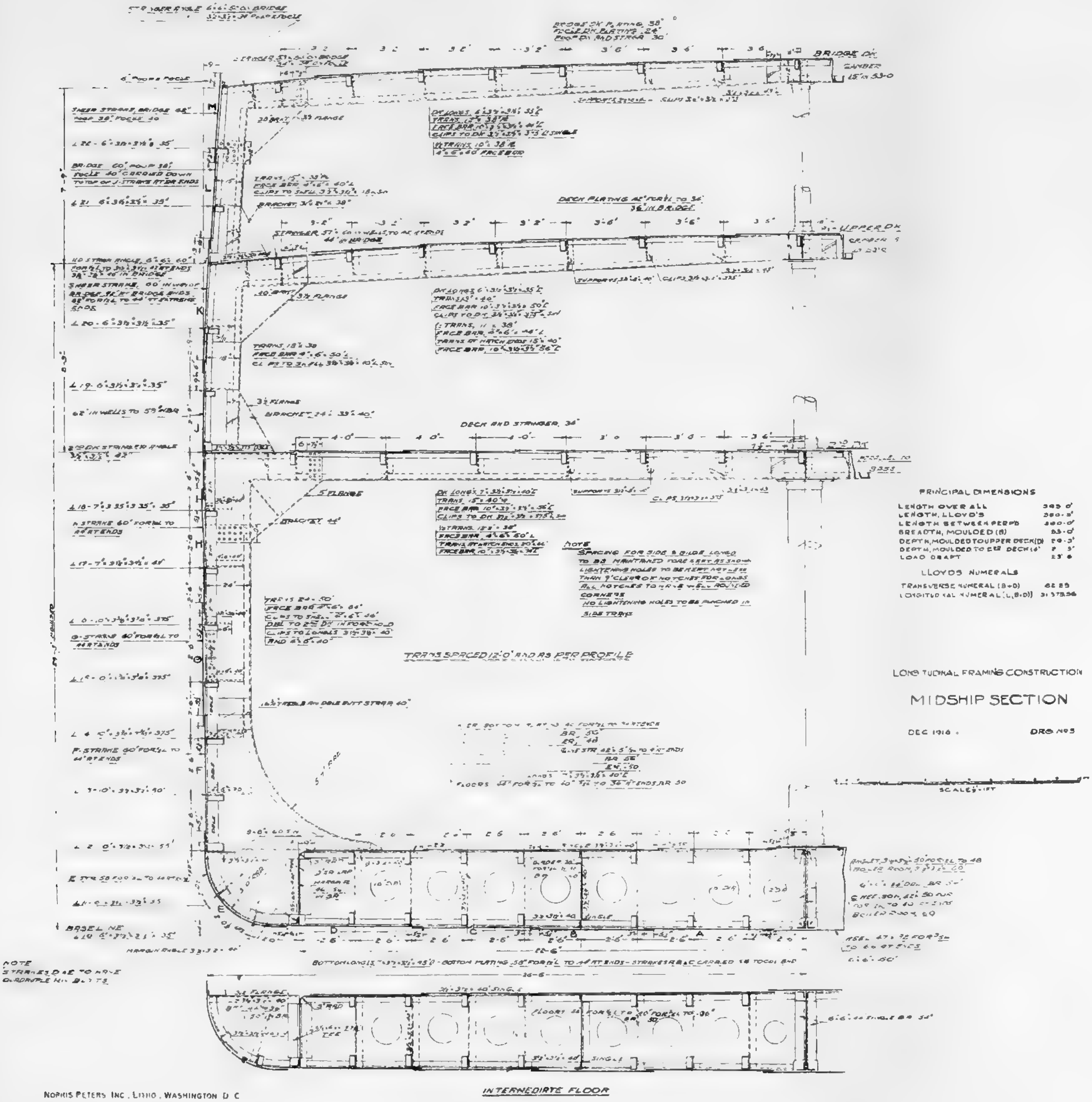

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Transactions Society Naval Architects and Marine Engineers, Vol. 27, 1919.

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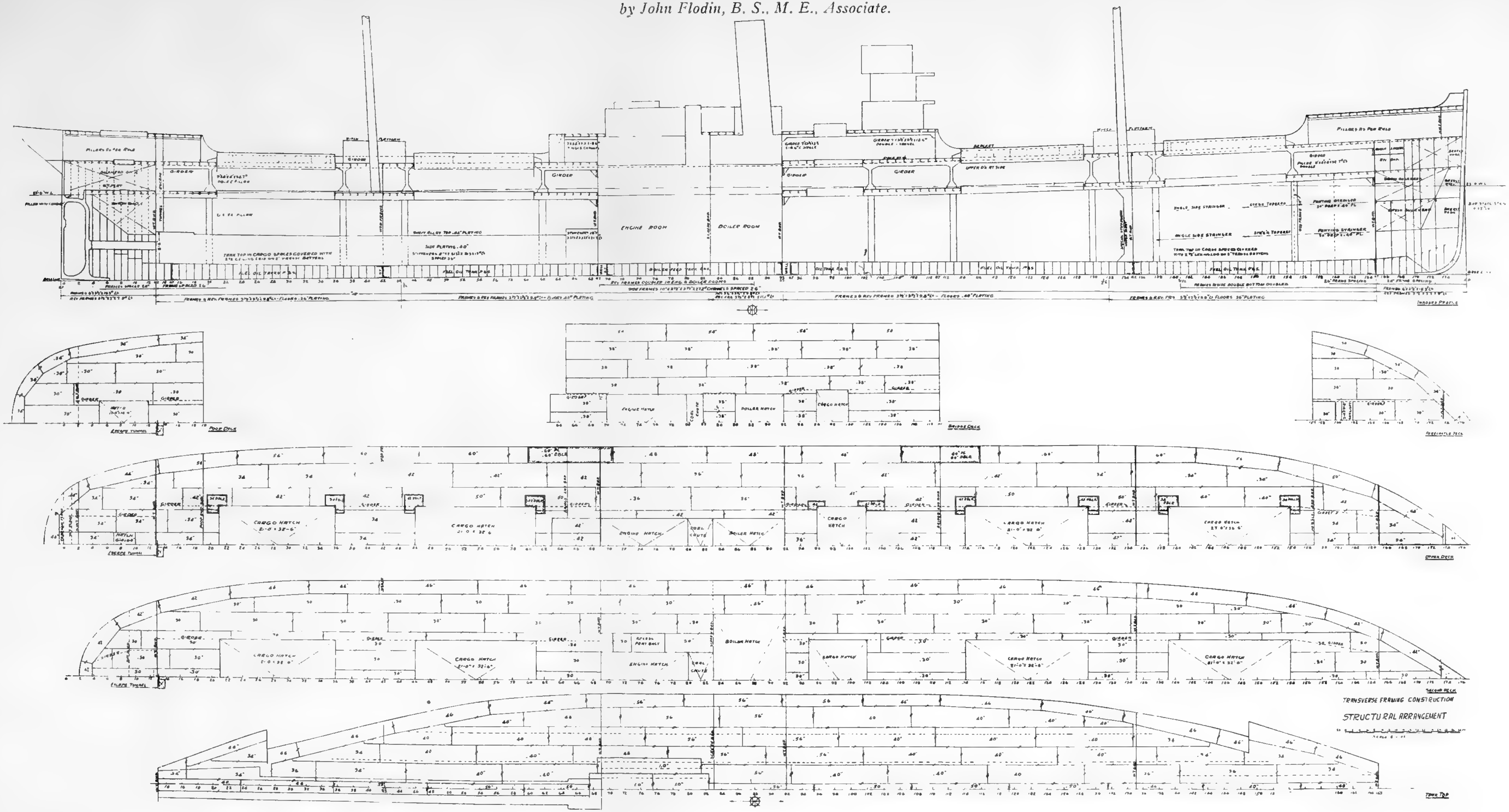

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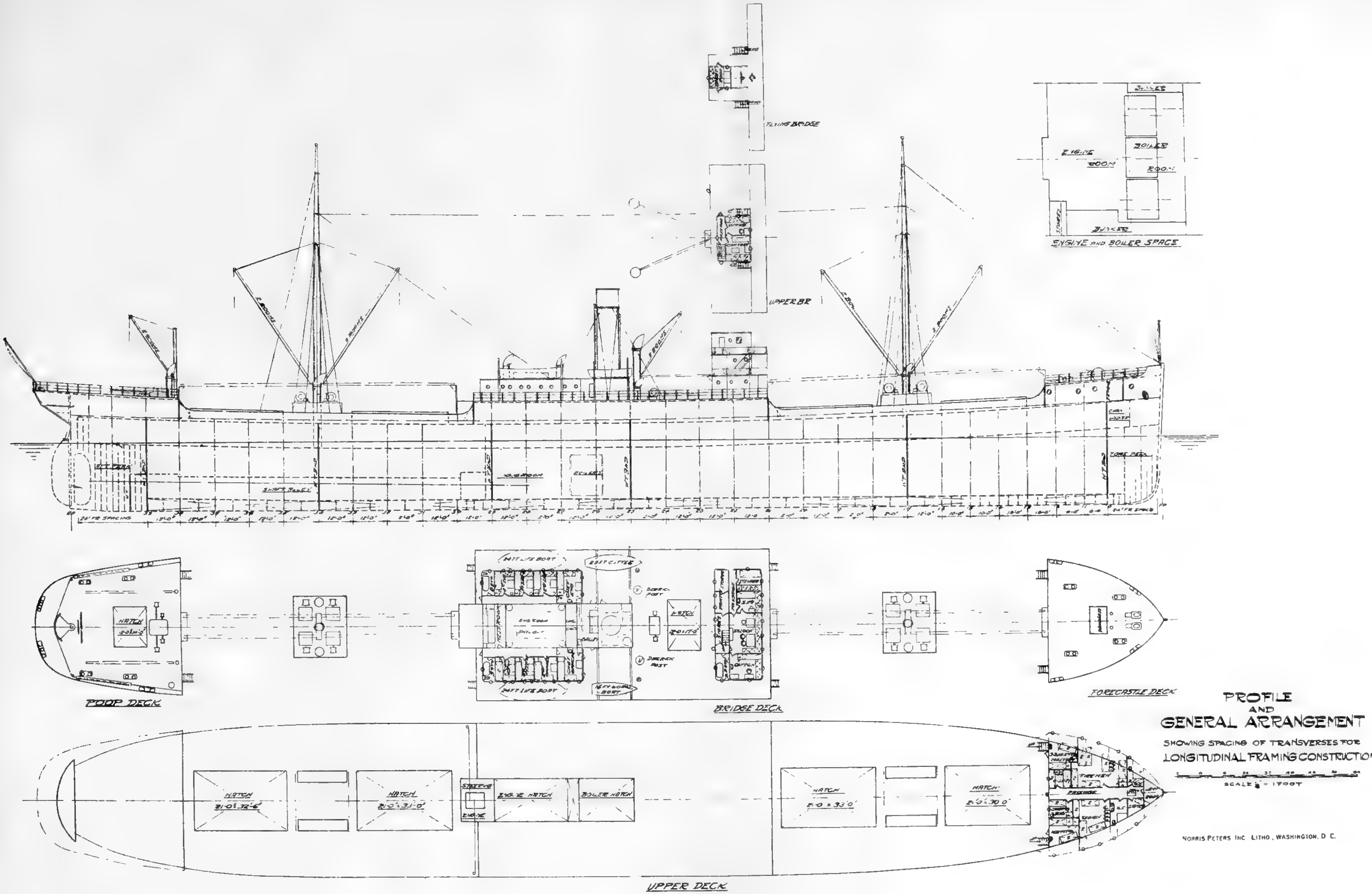

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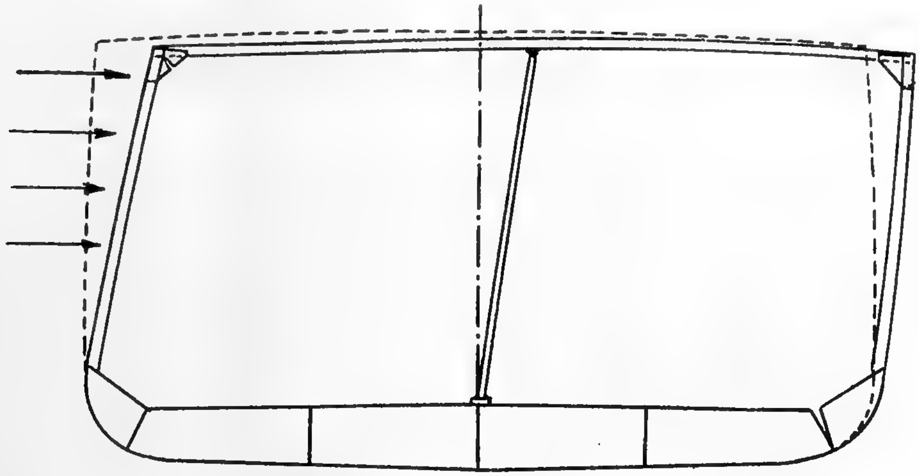

Fig. 1.

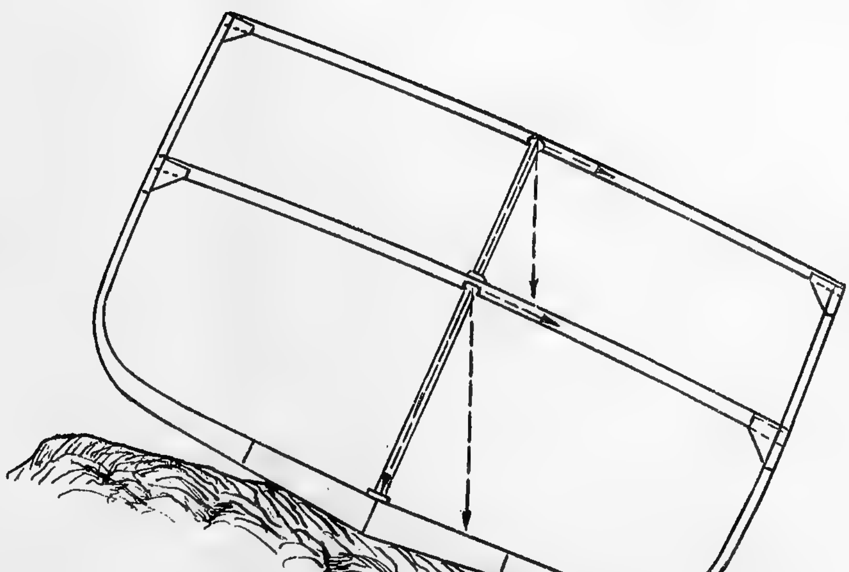

Fig 2.

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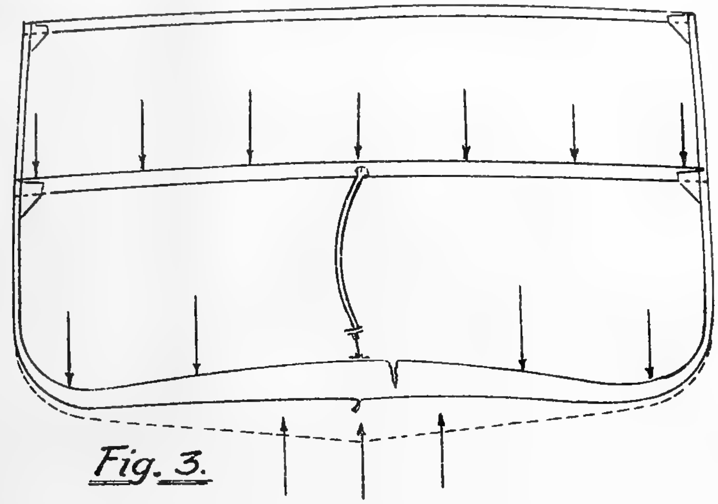

Fig. 3.

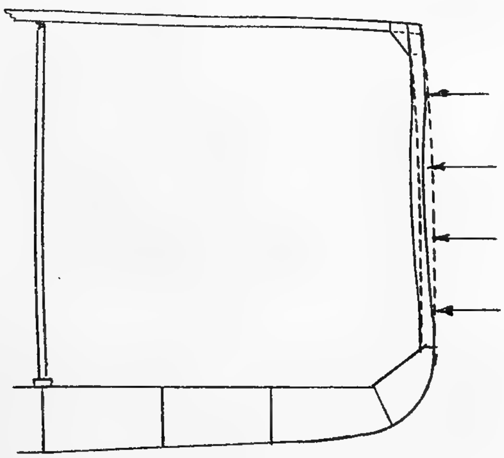

Fig. 4.

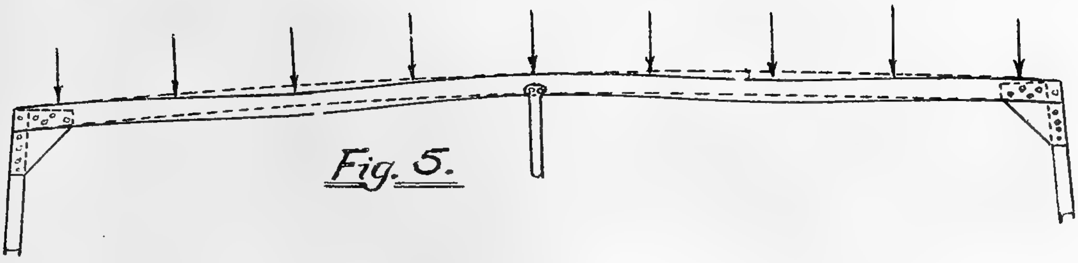

Fig. 5.

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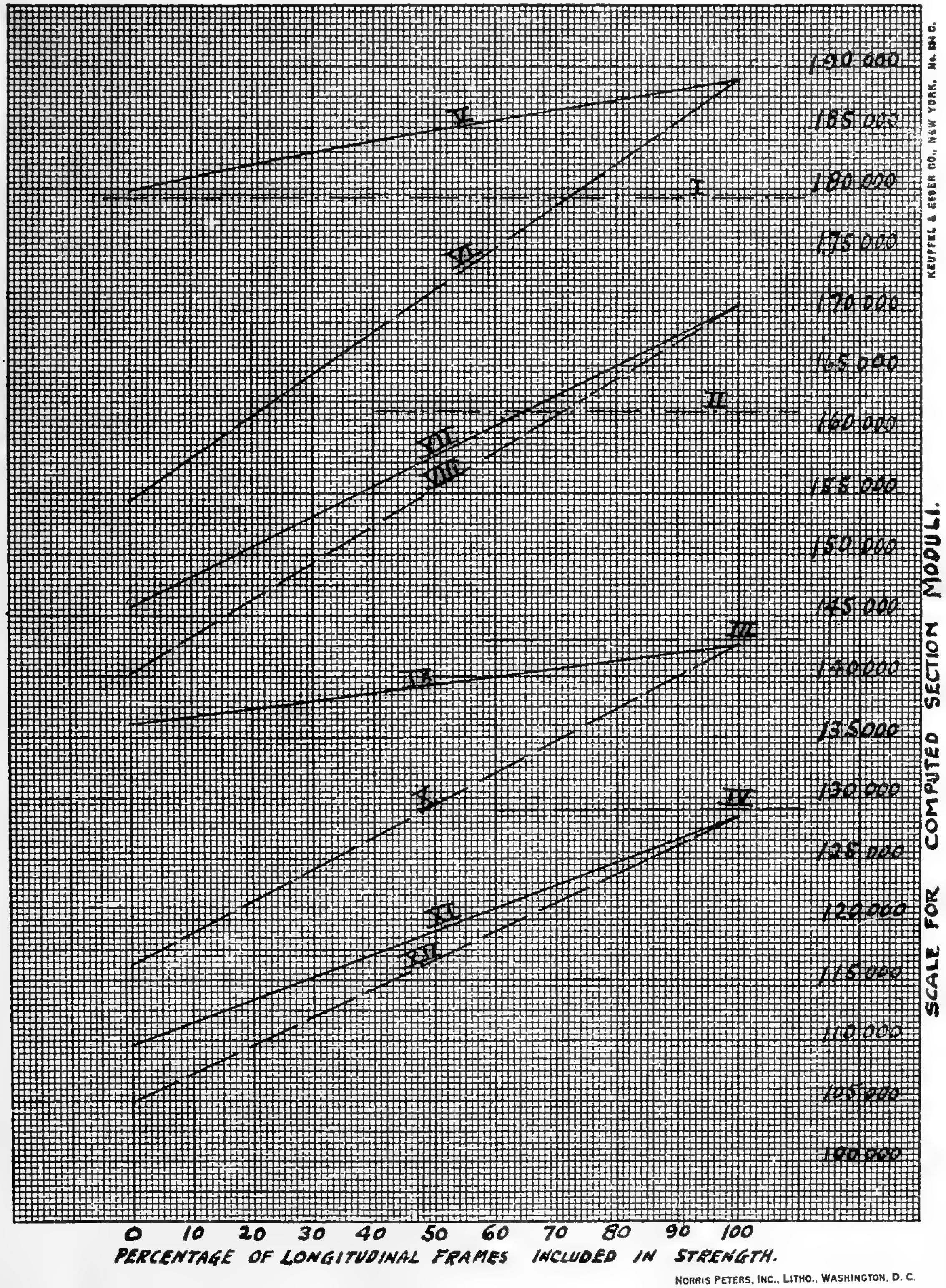
To illustrate discussion by F. M. Hiatt, Esq., Member, on paper entitled "An Analysis of the Isherwood System of Ship Construction," by John Flodin, B. S., M. E., Associate.



NORRIS PETERS, INC., LITHO., WASHINGTON, D. C.

To illustrate paper on "Economical Cargo Ships," by Alfred J. C. Robertson, Esq., Member.

No.	Column No. 2	3	350-Foot Ships, Minimum Port Detention 4	5	6	7	350 Foot Ships Double Port Detention 8	9	10	11
2	Length on waterline	Feet	350 0	350 0	350 0	350 0	350 0	350 0	350 0	350 0
3	Breadth		49 0	49 0	49 0	49 0	49 0	49 0	49 0	49 0
4	Load draft		21 9½	21 9½	21 9½	21 9½	21 9½	21 9½	21 9½	21 9½
5										
6	Speed on service	Knots	10 0	11 0	12 0	13 0	10 0	11 0	12 0	13 0
7	Speed length ratio	$\frac{V}{\sqrt{L}}$	535	587	641	695	535	587	641	695
8	Block coefficient	C_B	768	740	712	685	768	740	712	685
9	Load displacement	Tons	8 156	7 897	7 599	7 353	8 156	7 897	7 599	7 353
10	Indicated horse power	I.H.P.	1,566	1 970	2 470	3 028	1 566	1 970	2 470	3 028
11	Ratio tons displ. per I.H.P.	Tons	5 234	4 008	3 076	2 428	5 234	4 008	3 076	2 428
12										
13	Weight of hull and outfit	Tons	2 108	2 081	2 055	2 034	2 108	2 081	2 055	2 034
14	Weight of propelling mach'y		392	492	618	767	392	492	618	767
15	Total deadweight		5 696	5 324	4 926	4 562	5 696	5 324	4 926	4 562
16	Ratio deadweight to displ.		695	675	648	620	695	675	648	620
17										
18	Cost of hull and outfit	$	580 750	574 000	567 500	562 250	580 750	574 000	567 500	562 250
19	" " propelling machinery		203 580	256 100	321 100	393 640	203 580	256 100	321 100	393 640
20	" " ship		784 330	830 100	888 600	955 890	784 330	830 100	888 600	955 890
21	" " " per ton deadwt.		137 75	155 92	180 39	209 53	137 75	155 92	180 39	209 53
22										
23	Fuel consumed per voyage	Tons	265	300	342	384	285	320	362	404
24	Stores and fresh water		132	150	171	192	142	160	181	202
25	Cargo deadweight		5299	4874	4413	3986	5269	4844	4383	3956
26										
27	Time on voyage	Days	14 58	13 26	12 15	11 22	14 58	13 26	12 15	11 22
28	" in port		5 15	5 15	5 15	5 15	6 30	10 3	10 30	10 30
29	" for annual overhaul		30 00	30 00	30 00	30 00	30 00	30 00	30 00	30 00
30										
31	Number of trips per year		16 57	18 20	19 36	20 46	13 46	14 2	14 92	15 67
32	Time on voyage per year	Days	247 4	241 3	235 2	229 6	196 4	188 3	181 3	174 6
33	Time in port per year		117 6	123 7	129 8	135 4	168 6	176 5	183 7	190 4
34	Percentage time under way	%	67 8	66 1	64 4	62 9	53 8	51 6	49 7	47 8
35										
36	Cost of manning and manage't p.a.	$	72 915	72 915	72 915	72 915	72 915	72 915	72 915	72 915
37	Depreciation insurance upkeep p.a.		101 963	107 9[illegible]3	115 518	124 266	101 963	107 9[illegible]3	115 518	124 266
38	Fuel oil @ $6.00 a ton per annum		26 988	32 760	39 777	47.141	23 017	27 307	32 466	37 742
39	Dues @ $1.50 per nett ton p.a.		65 545	65 474	73 549	75 211	55 559	56 [illegible]	56 681	57 235
40										
41	Annual / Fuel @ $6.00 no dues	$	201 866	213 588	228160	244 322	197 895	208 1[illegible]	220 839	234 922
42	cost of / " with dues		271 409	285 442	301 709	319 533	253 054	264 3[illegible]	277 520	292 158
43	operation / " @ $12.00 "		298 897	318 202	341 496	366 674	276 071	29[illegible]	309 926	329 900
44										
45	Cargo moved p.a. per $1000000	Tons	114 650	106 863	96 146	81 553	90 422	82 91[illegible]	73 592	64 437
46										
47	Cost of operating ($6 fuel no dues)	$	257 374	267 364	256 763	255 696	252 311	250 7[illegible]	246 525	245 763
48	per annum per million ($6 with dues)		344 059	343 864	355 553	334 778	322 637	318 3[illegible]	312 311	305640
49	dollars investment ($12 ")		386 440	383 329	384 240	393 394	351 983	351 2[illegible]	348 780	345 173
50										
51	Profit p.a. per $100000 ($6 fuel no dues)	$	198 478	266 981	209 545	139 936	186 236	151 7[illegible]	108 396	66 756
52	investment @ $6.00 per ($6 with dues)		210 013	174 421	126,775	61 254	115 510	84 076	44 610	6 879
53	ton, less $1.00 handl'g & comm'n ($12 ")		175 612	154 556	82 048	11 538	86 364	51 1[illegible]	8 41[illegible]	Loss 32 604
54										
55										

No.	Column No. 2	3	400-Foot Ships, Minimum Port Detent. 12	13	14	15	450-Foot Ships Minimum Detention 16	17	18	19
2	Length on waterline	Feet	400 0	400 0	400 0	400 0	450 0	450 0	450 0	450 0
3	Breadth		54 0	54 0	54 0	54 0	59 0	59 0	59 0	59 0
4	Load draft		24 0	24 0	24 0	24 0	26 2½	26 2½	26 2½	26 2½
5										
6	Speed on service	Knots	10 0	11 0	12 0	13 0	10 0	11 0	12 0	13 0
7	Speed length ratio	$\frac{V}{\sqrt{L}}$	50	55	60	65	471	518	565	613
8	Block coefficient	C_B	789	762	734	711	809	782	755	731
9	Load displacement	Tons	11 686	11 286	10 872	10 531	16 090	5 553	15 016	14 539
10	Indicated horse power	I.H.P.	2 021	2 515	3 121	3 862	2 600	3 200	3 877	4 749
11	Ratio tons displ. per I.H.P.	Tons	5 782	4 487	3 483	2 727	6 188	4 860	3 873	3 061
12										
13	Weight of hull and outfit	Tons	3015	2978	2941	2909	4 166	4 119	4 069	4 026
14	Weight of propelling mach'y		505	629	780	966	650	800	969	1 187
15	Total deadweight		8166	7679	7151	6656	11 274	10 634	9 978	9 326
16	Ratio deadweight to displ.		699	680	658	632	701	684	664	641
17										
18	Cost of hull and outfit	$	833 750	824 500	816 250	807 250	1 155 750	1 143 500	1 131,000	1 120 250
19	" " propelling machinery		262 730	326 950	405 730	502 060	338 000	416 000	504 010	617 370
20	" " ship		1 096 480	1 151 450	1 220 980	1 309 310	1 493 750	1 559 500	1 635 000	1 737 620
21	" " " per ton deadwt.		134 27	149 95	170 74	196 71	132 46	146 65	163 85	186 32
22										
23	Fuel consumed per voyage	Tons	346	387	436	494	446	494	545	611
24	Stores and fresh water		173	193	218	247	223	247	272	305
25	Cargo deadweight		7647	7 099	6 497	5 915	10 605	9893	9 161	8 410
26										
27	Time on voyage	Days	14 58	13 26	12 15	11 22	14 58	13 26	12 15	11 22
28	" in port		5 80	5 80	5 80	5 80	6 50	6 50	6 50	6 50
29	" for annual overhaul		30 00	30 00	30 00	30 00	30 00	30 00	30 00	30 00
30										
31	Number of trips per year		16 44	17 58	18 66	19 68	15 89	16 95	17 96	18 90
32	Time on voyage per year	Days	239 7	233 1	226 7	220 8	231 7	224 8	218 2	217 1
33	Time in port per year		125 3	131 9	138 3	44 2	133 3	140 2	46 8	152 9
34	Percentage time under way	%	65 7	63 9	62 1	60 5	63 5	61 6	59 8	58 1
35										
36	Cost of manning and manage't p.a.	$	81 300	81 300	81 300	81 300	98 490	98 490	98 490	98 490
37	Depreciation insurance upkeep p.a.		142 542	149 688	158 727	170 210	194 122	202 735	212 550	225,891
38	Fuel oil @ $6.00 a ton per annum		34 150	40 862	48815	58 332	42 322	50,241	58 729	69 290
39	Dues @ $1.50 per nett ton p.a.		94 039	99 104	101 456	103 6[illegible]5	27 835	15 803	134 844	137 384
40										
41	Annual / Fuel @ $6.00 no dues	$	237972	271 810	288 842	309 842	335 134	351 466	369 769	393 671
42	cost of / " with dues		334 031	371 014	390 278	413,457	462 969	483 269	504 613	531,055
43	operation / " @ $12.00 "		358 161	[illegible]	430 093	47[illegible] 789	565 491	533 5[illegible]0	563 342	600 345
44										
45	Cargo moved p.a. per $1000000	Tons	116 655	108 385	99 344	88 907	112 835	107 525	100,742	91 475
46										
47	Cost of operating ($6 fuel no dues)	$	235 273	236 059	236 565	236 645	224 433	225 371	226 5[illegible]	226 557
48	per annum per million ($6 with dues)		322 879	322 215	319 643	315 782	310 041	309 887	308 632	305 622
49	dollars investment ($12 ")		354 006	357 667	359 623	36[illegible] 334	338 5[illegible]7	342 103	344 551	345 498
50										
51	Profit p.a. per $100000 ($6 fuel no dues)	$	320 804	289 608	245 253	194 554	322 817	296 125	262 441	217 097
52	investment @ $6.00 per ($6 with dues)		233 158	203 452	162 175	115 417	237,709	211,609	179 967	138 032
53	ton, less $1.00 handl'g & comm'n ($12 ")		207 071	168 000	122 195	70 965	208 753	179 393	144,048	98 156
54										
55										

No.	Column No. 2	3	500 Foot Ships, Minimum Detention 20	21	22	23	500 Foot Ships, Double Detention 24	25	26	27
2	Length on waterline	Feet	500 0	500 0	500 0	500 0	500 0	500 0	500 0	500 0
3	Breadth		64 0	64 0	64 0	64 0	64 0	64 0	64 0	64 0
4	Load draft		28 5¼	28 5¼	28 5¼	28 5¼	28 5¼	28 5¼	28 5¼	28 5¼
5										
6	Speed on service	Knots	10 0	11 0	12 0	13 0	10 0	11 0	12 0	13 0
7	Speed length ratio	$\frac{V}{\sqrt{L}}$	447	491	536	581	447	491	536	581
8	Block coefficient	C_B	825	80	775	750	825	80	775	750
9	Load displacement	Tons	21 452	20 802	20,152	19 500	21 452	20 802	20 152	19 500
10	Indicated horse power	I.H.P.	3 254	3 994	4 825	5 828	3 254	3 994	4 825	5 828
11	Ratio tons displ. per I.H.P.	Tons	6 592	5 208	4 176	3 345	6 592	5 208	4 176	3 345
12										
13	Weight of hull and outfit	Tons	5 626	5 568	5 506	5 448	5 626	5 568	5 506	5448
14	Weight of propelling mach'y		814	998	1 206	1 457	814	998	1 206	1 457
15	Total deadweight		15 012	14 236	13 440	12 595	15 012	14 236	13 440	12 595
16	Ratio deadweight to displ.		700	684	666	646	700	684	666	646
17										
18	Cost of hull and outfit	$	1 562 750	1 548,250	1 532 750	1 518,250	1 562 750	1 548 250	1 532 750	1 518 250
19	" " propelling machinery		423 020	519 270	627 250	757 640	423 020	519 220	627 250	757 640
20	" " ship		1 985 770	2 067 470	2,160 000	2 275 890	1 985 770	2 067 470	2 160 000	2 275 890
21	" " " per ton deadwt.		132 27	145 72	160 71	180 67	132 27	145 72	160 71	180 67
22										
23	Fuel consumed per voyage	Tons	558	620	678	751	585	670	728	801
24	Stores and fresh water		278	310	339	375	291	335	364	400
25	Cargo deadweight		14 175	13 306	12 423	11 469	14 138	13 231	12 348	11 394
26										
27	Time on voyage	Days	14 58	13 26	12 15	11 22	14 58	13 26	12 15	11 22
28	" in port		7 25	7 25	7 25	7 25	14 50	14 50	14 50	14 50
29	" for annual overhaul		30 00	30 00	30 00	30 00	30 00	30 00	30 00	30 00
30										
31	Number of trips per year		15 34	16 33	17 26	18 14	11 52	12 07	12 57	13 02
32	Time on voyage per year	Days	223 7	216 5	209 7	203 5	168 0	160 0	152 7	146 2
33	Time in port per year		141 3	148 5	156 3	161 5	197 0	205 0	212 3	2 8 8
34	Percentage time under way	%	61 3	59 3	57 5	55 7	46 0	43 8	41 8	40 0
35										
36	Cost of manning and manage't p.a.	$	106 176	106 176	106 176	106 176	106 176	106 176	106 176	106 176
37	Depreciation insurance upkeep p.a.		258,150	268 771	280 800	295 865	258 150	268 771	280 800	295 865
38	Fuel oil @ $6.00 a ton per annum		51,358	60 750	70,214	81 739	40 297	48 521	54 906	62 574
39	Dues @ $1.50 per nett ton p.a.		156 385	169 848	173 512	176 865	123 563	125 540	126 655	126 945
40										
41	Annual / Fuel @ $6.00 no dues	$	415 684	435,697	457 190	483780	404 623	473 468	441 882	464 615
42	cost of / " with dues		572,069	605 545	631 102	660645	528 186	549 008	568 537	591 560
43	operation / " @ $12.00 "		623 427	666 295	701 3[illegible]	742 384	568 483	597 129	623 445	654 134
44										
45	Cargo moved p.a. per $1000000	Tons	105 501	106 098	99 269	91 414	82 018	77 245	71 858	65 183
46										
47	Cost of operating ($6 fuel no dues)	$	209 331	210 739	211 662	212 568	263 761	204 824	204 575	204 4[illegible]
48	per annum per million ($6 with dues)		288 084	292 891	292 176	290 280	265 585	265 546	263 211	259 925
49	dollars investment ($12 ")		315 947	322 275	324 683	326 196	286 278	285 015	288 631	287 4[illegible]9
50										
51	Profit p.a. per $100000 ($6 fuel no dues)	$	371 749	298 986	269 793	230 790	194 018	169 804	143 936	111 991
52	investment @ $6.00 per ($6 with dues)		242 596	216 834	189 279	153 078	151 802	109 082	85 300	56 2[illegible]2
53	ton, less $1.00 handl'g & comm'n ($12 ")		217 133	187 450	66 772	117 162	11[illegible] 509	85 613	53 880	28 7'8
54										
55										

No.	Column No. 2	3	500 Foot Ships 10% Greater Power 28	29	30	31	500 Foot Ship Block Coeff. 750 32	33	34	35
2	Length on waterline	Feet	500 0	500 0	500 0	500 0	500 0	500 0	500 0	500 0
3	Breadth		64 0	64 0	64 0	64 0	64 0	64 0	64 0	64 0
4	Load draft		28 5¼	28 5¼	28 5¼	28 5¼	28 5¼	28 5¼	28 5¼	28 5¼
5										
6	Speed on service	Knots	10 0	11 0	12 0	13 0	10 0	11 0	12 0	13 0
7	Speed length ratio	$\frac{V}{\sqrt{L}}$	447	491	536	581	447	491	536	581
8	Block coefficient	C_B	825	80	775	750	750	750	750	750
9	Load displacement	Tons	21 452	20 802	20 152	19 500	9 500	19 500	19 500	19 500
10	Indicated horse power	I.H.P.	3 579	4 393	5 308	6 411	2 582	3 460	4 519	5 828
11	Ratio tons displ. per I.H.P.	Tons	5 993	4 735	3 797	3 041	7 552	5 636	4 315	3 346
12										
13	Weight of hull and outfit	Tons	5 626	5 568	5 506	5 448	5 448	5 448	5 448	5 448
14	Weight of propelling mach'y		895	1 098	1 327	1 603	1 457	1 457	1 457	1,457
15	Total deadweight		4 931	14 136	13 319	12 449	12 595	2 595	12 595	12 595
16	Ratio deadweight to displ.						646	646	646	646
17										
18	Cost of hull and outfit	$	1 562 750	1 548 25[illegible]	1 532 750	1 518 250	1 518 250	1 518 250	1 518 250	1 518 250
19	" " propelling machinery		465 323	571 142	689 975	833 404	757 640	757 640	757 640	757 640
20	" " ship		2 028 073	2 1[illegible] 39[illegible]	2 222 725	2 35[illegible] 654	2 275 890	2 275 890	2 275 890	2 275 890
21	" " " per ton deadwt.		135 [illegible]	149 92	166 [illegible]	188 [illegible]	180 47	180 47	180 47	180 47
22										
23	Fuel consumed per voyage	Tons	614	682	746	826	454	543	640	751
24	Stores and fresh water		307	341	373	413	227	271	320	375
25	Cargo deadweight		14 010	13 1[illegible]3	12 200	11 210	11 914	11 781	11 635	11 469
26										
27	Time on voyage	Days	14 58	13 26	12 15	11 22	14 58	13 26	12 5	11 22
28	" in port		7 25	7 25	7 25	7 25	7 25	7 25	7 25	7 25
29	" for annual overhaul		30 00	30 00	30 00	30 00	30 00	30 00	30 00	30 00
30										
31	Number of trips per year		15 34	16 33	17 26	18 14	15 34	16 33	17 26	18 14
32	Time on voyage per year	Days	223 7	216 5	209 7	203 5	223 7	216 5	209 7	203 5
33	Time in port per year		14[illegible] 5	148 5	155 3	16[illegible] 5	141 3	148 5	155 3	161 5
34	Percentage time under way	%	6[illegible] 3	59 3	57 5	55 7	61 3	59 3	57 5	55 7
35										
36	Cost of manning and manage't p.a.	$	106 176	106 176	106 176	106 176	106 176	106 176	106 176	106 176
37	Depreciation insurance upkeep p.a.		263 649	275 [illegible]	288 554	305 715	295 865	295 865	295 865	295 865
38	Fuel oil @ $6.00 a ton per annum		53 712	66 822	77 255	89 513	41 786	53 205	66 278	81,739
39	Dues @ $1.50 per nett ton p.a.		156 385	169 8[illegible]	173 9[illegible]2	176 865	149 56[illegible]	159 2[illegible]	168 185	176 865
40										
41	Annual / Fuel @ $6.00 no dues	$	423 537	448 5[illegible]9	472 365	501 804	443 827	455 244	468 319	483,780
42	cost of / " with dues		579 922	618 367	646 277	678 669	593 392	614 461	636 604	660645
43	operation / " @ $12.00 "		633 634	68[illegible]	725 612	768 582	635 [illegible]	667 664	7[illegible]2 882	742 384
44										
45	Cargo moved p.a. per $1000000	Tons	105 569	101 036	94 736	86 471	80 305	84 531	88 238	91,414
46										
47	Cost of operating ($6 fuel no dues)	$	208 857	[illegible]	212 5[illegible]	213 383	19[illegible]	200 030	205 775	212 568
48	per annum per million ($6 with dues)		285 947	291 766	290 759	288 592	260 750	269 988	279 717	290 280
49	dollars investment ($12 ")		3[illegible]2 431	323 2[illegible]	326 307	326 826	273 051	293 365	308 839	326196
50										
51	Profit p.a. per $100000 ($6 fuel no dues)	$	305 113	276 398	246 954	206 001	194,457	209 945	227 179	230 790
52	investment @ $6.00 per ($6 with dues)		228 003	198 255	168 711	130 792	28 740	139 987	148,737	153 078
53	ton, less $1.00 handl'g & comm'n ($12 ")		201 519	66 73	133 963	92 558	110 379	1[illegible]6 610	119 115	117 162
54										
55										

No.	Column No. 2	3	500 Foot Ship, Block Coeff. 825 36	37	38	39
2	Length on waterline	Feet	500 0	500 0	500 0	500 0
3	Breadth		64 0	64 0	64 0	64 0
4	Load draft		28 5¼	28 5¼	28 5¼	28 5¼
5						
6	Speed on service	Knots	10 0	11 0	12 0	13 0
7	Speed length ratio	$\frac{V}{\sqrt{L}}$	447	491	536	581
8	Block coefficient	C_B	825	825	825	825
9	Load displacement	Tons	[illegible] 452	2[illegible] 452	21 452	21 452
10	Indicated horse power	I.H.P.	3 254	4 032	5 935	8 2[illegible]5
11	Ratio tons displ. per I.H.P.	Tons	6 592	5 320	3 614	2 611
12						
13	Weight of hull and outfit	Tons	[illegible]	5 626	5 626	5 626
14	Weight of propelling mach'y		2 054	2 054	2 054	2 054
15	Total deadweight		[illegible]	13 772	13 772	3 772
16	Ratio deadweight to displ.		[illegible]	642	642	642
17						
18	Cost of hull and outfit	$	[illegible] 750	1 562 750	1 562 750	1 562 750
19	" " propelling machinery		[illegible] 350	[illegible]	067 950	067 950
20	" " ship		[illegible]	2 630 700	2 630 700	2 630 700
21	" " " per ton deadwt.		191 02	191 02	191 02	191 02
22						
23	Fuel consumed per voyage	Tons	[illegible]	623	824	1038
24	Stores and fresh water		[illegible]	312	4[illegible]2	519
25	Cargo deadweight		[illegible]	2 837	2 53[illegible]	12 215
26						
27	Time on voyage	Days	[illegible]	13 26	12 5	11 22
28	" in port		7 25	7 25	7 25	7 25
29	" for annual overhaul		[illegible] 00	30 00	30 00	30 00
30						
31	Number of trips per year		[illegible] 34	[illegible] 33	[illegible]	18 14
32	Time on voyage per year	Days	[illegible]	216 5	[illegible]	203 5
33	Time in port per year		[illegible]	148 5	155 5	161 5
34	Percentage time under way	%	[illegible]	59 3	57 5	55 7
35						
36	Cost of manning and manage't p.a.	$	[illegible]	[illegible]	[illegible]	106 176
37	Depreciation insurance upkeep p.a.		[illegible]	[illegible]	34 99	34 991
38	Fuel oil @ $6.00 a ton per annum		51 450	61 041	85 333	112 976
39	Dues @ $1.50 per nett ton p.a.		[illegible]	175 172	85 148	194 588
40						
41	Annual / Fuel @ $6.00 no dues	$	[illegible] 617	509 208	[illegible]	[illegible] 43
42	cost of / " with dues		564 169	684 380	7 8648	755731
43	operation / " @ $12.00 "		[illegible]	745 42[illegible]	803 98	848 707
44						
45	Cargo moved p.a. per $1000000	Tons	74 420	79 685	82 248	94 228
46						
47	Cost of operating ($6 fuel no dues)	$	[illegible]	[illegible] 564	272 798	2[illegible]
48	per annum per million ($6 with dues)		252 469	260 151	273 177	28[illegible] 274
49	dollars investment ($12 ")		272 026	285355	305 615	330 [illegible]
50						
51	Profit p.a. per $100000 ($6 fuel no dues)	$	[illegible] 669	192 908	196 105	195 200
52	investment @ $6.00 per ($6 with dues)		[illegible]	26 32[illegible]	25 725	[illegible] 232
53	ton, less $1.00 handl'g & comm'n ($12 ")		[illegible]	3[illegible] 7	93 288	78 287
54						
55						

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To illustrate paper on "Economical Cargo Ships," by Alfred J. C. Robertson, Esq., Member.

Fig. 1

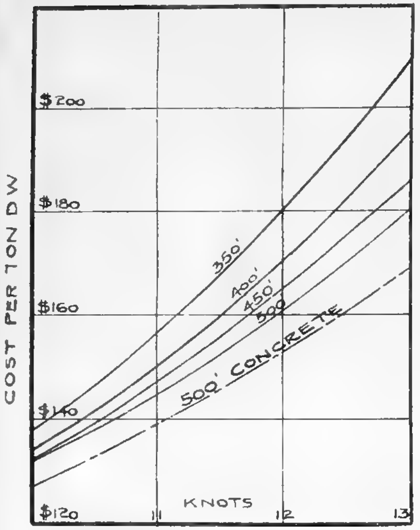
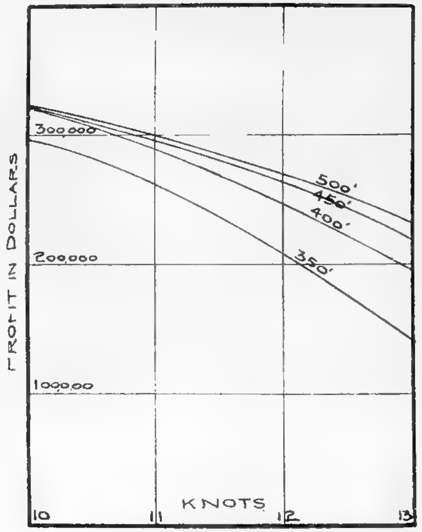
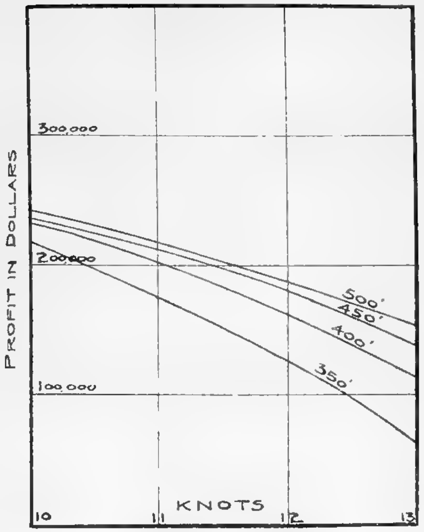


Fig. 2



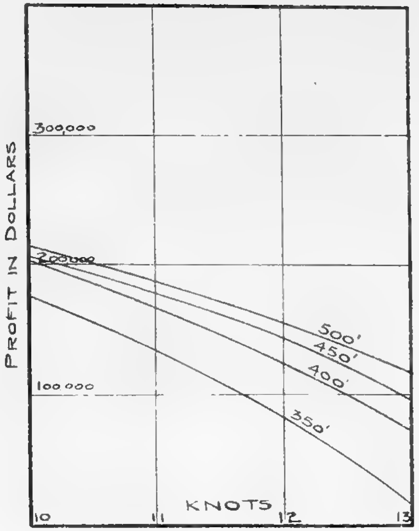
OIL FUEL @ $6.00 NO DUES

Fig. 3



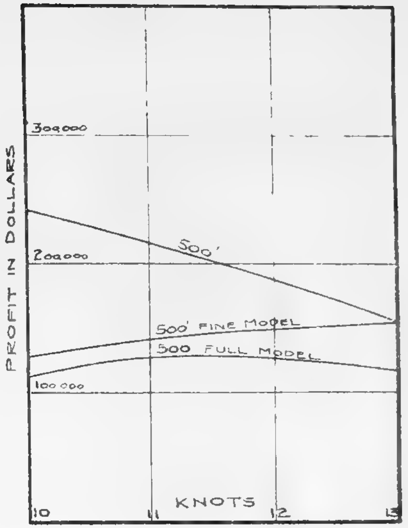
OIL FUEL @ $6.00, DUES 1.50

Fig. 4



OIL FUEL $12.00, DUES $1.50

Fig. 5



OIL FUEL $6.00 DUES $1.50

Fig. 6

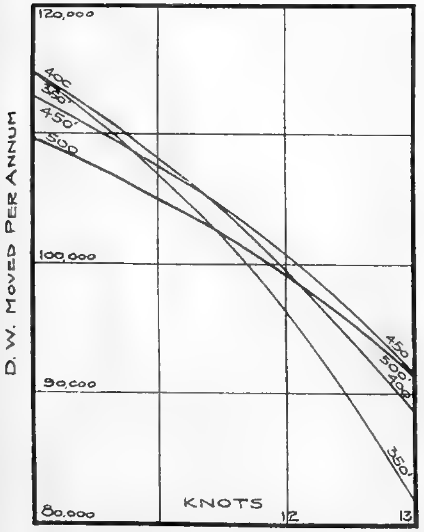


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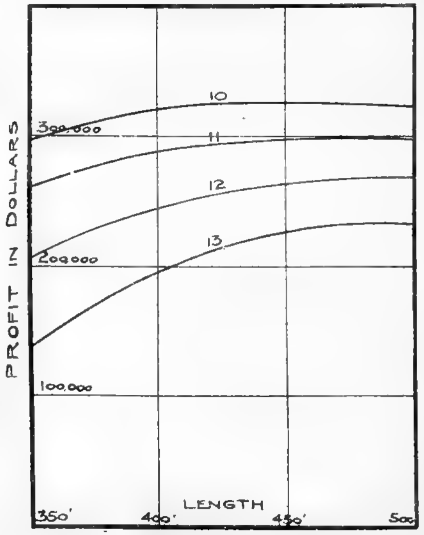
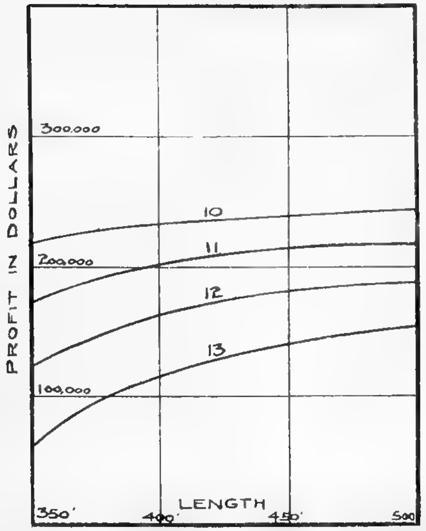
Fig. 7



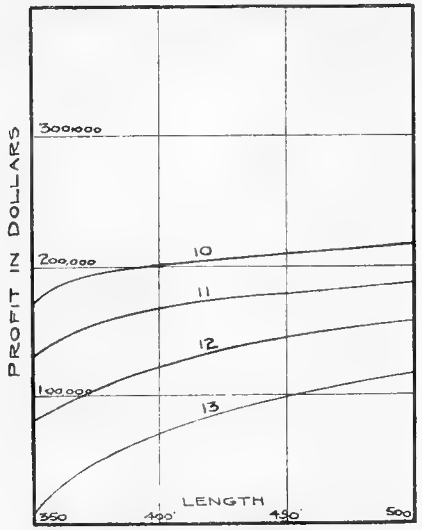
OIL FUEL $6.00 NO DUES

Fig. 8



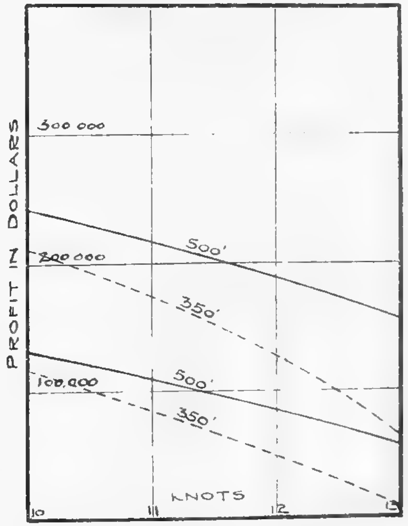
OIL FUEL $6.00, DUES $1.50

Fig. 9



OIL FUEL $12 DUES $1.50

Fig. 10.



DOUBLE DETENTION AT PORT WITH OIL FUEL $6.00 DUES $1.50

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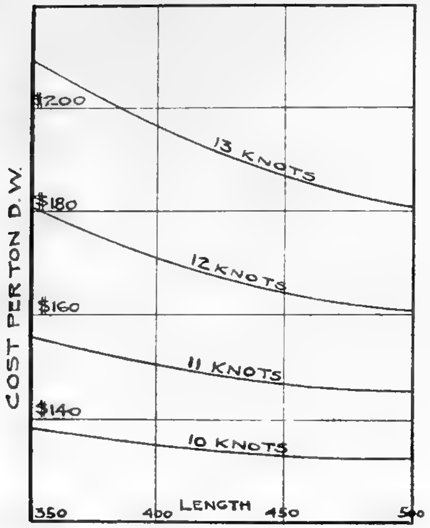
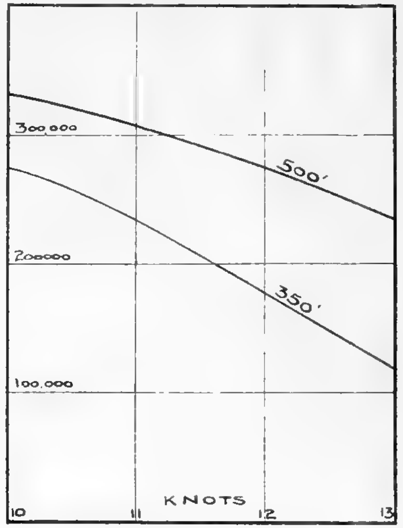
FIG 11.



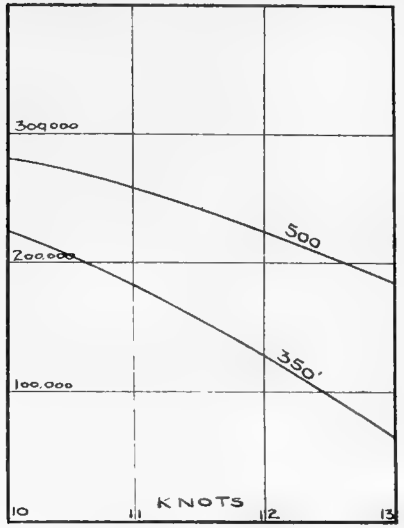
COST

FIG 12.



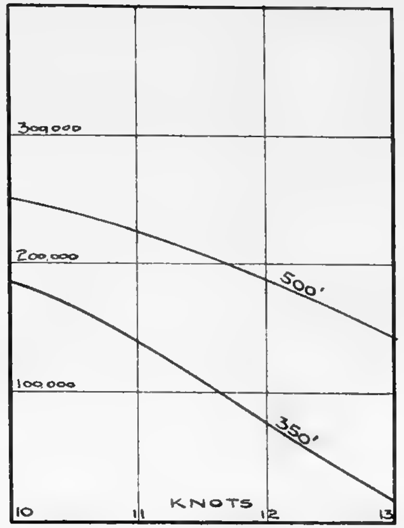
OIL FUEL @ $6.00 NO DUES
7000 KNOTS

FIG 13



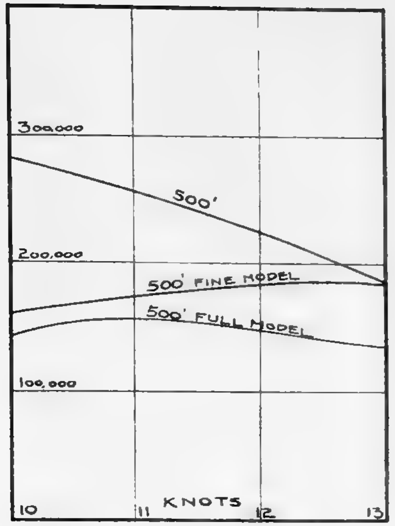
OIL FUEL @ $6.00 DUES $1.50
7000 KNOTS

FIG. 14



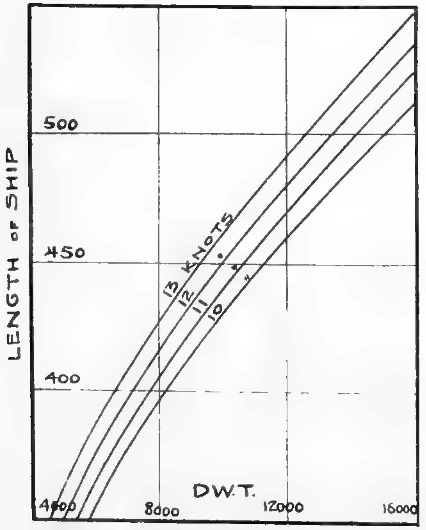
OIL FUEL $12.00, DUES $1.50
7000 KNOTS

FIG 15.



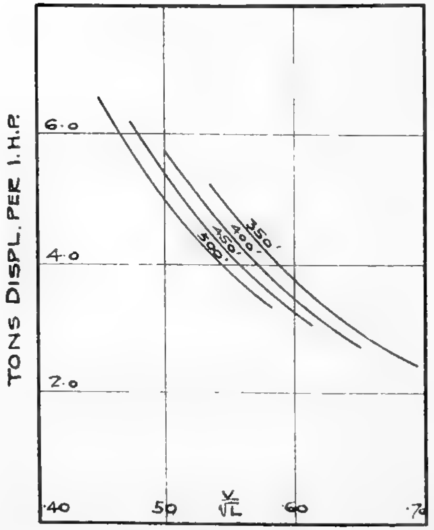
OIL FUEL $6.00 DUES $1.50
7000 KNOTS

FIG. 16



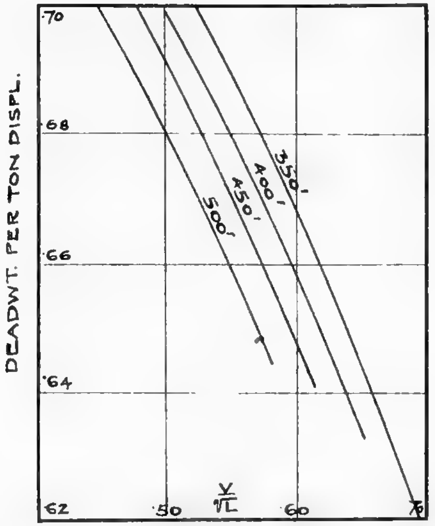
LENGTH

FIG. 17



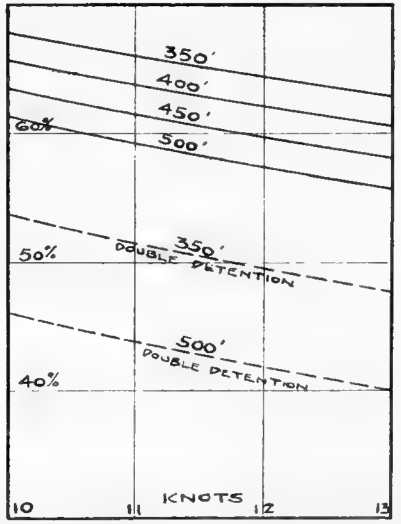
POWER

FIG. 18



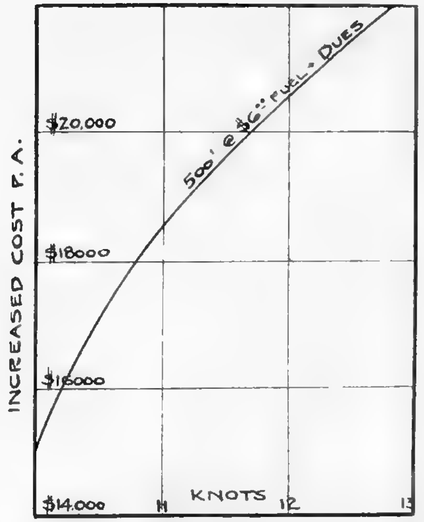
LOAD

FIG. 19



PER CENT RUNNING TIME

FIG. 20



10% GREATER POWER

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To illustrate paper on "Non-Rolling Passenger Liners—Observations on a Large Stabilized Ship in Service, Including the Plant and Economies Effected by Stabilization," by Elmer A. Sperry, Esq., Member.

Fig. 1.—Some of the Thousands of Feet of Records Used in Studying Ship Stabilization.

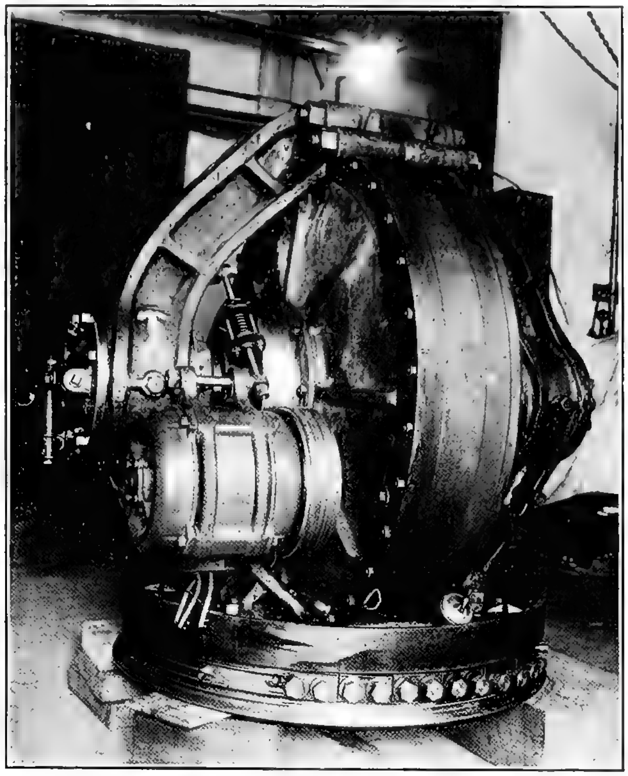

Fig. 2.—Original Active Stabilizer Installed on Destroyer Worden.

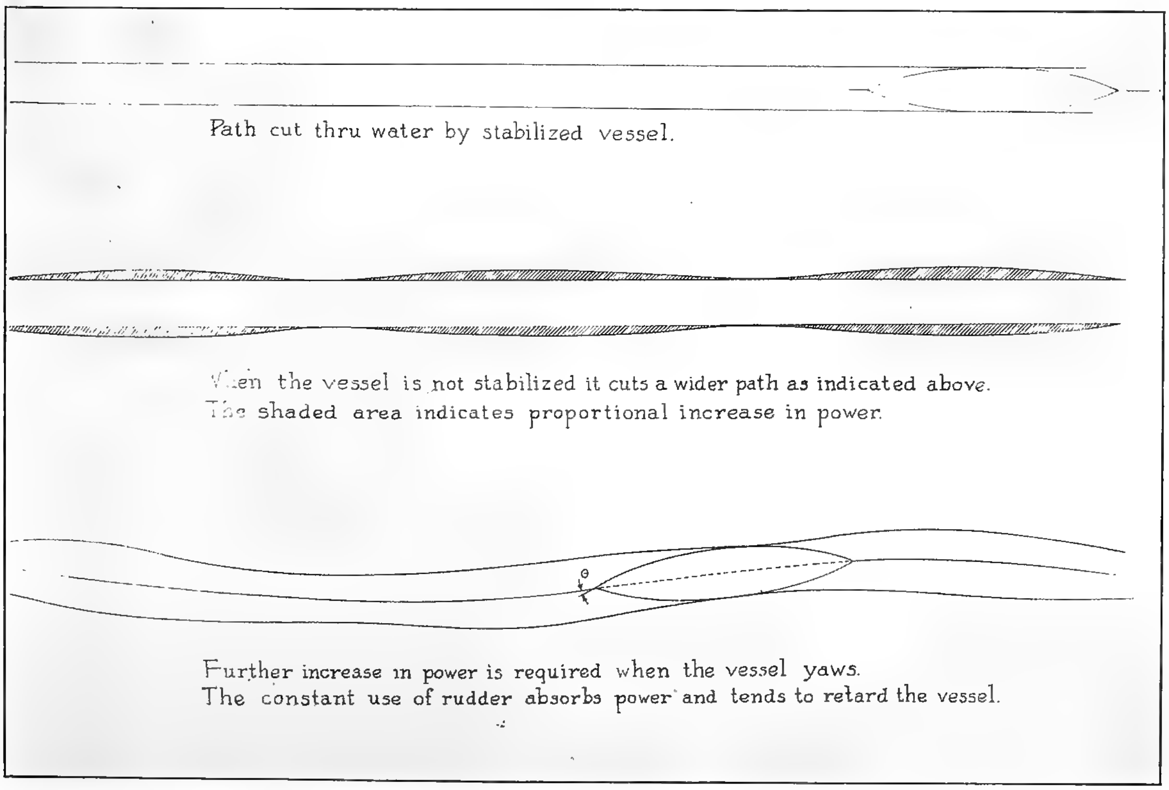

Fig. 3.—Losses Due to Helm, Rolling, and Sinuous Course.

To illustrate paper on "Non-Rolling Passenger Liners—Observations on a Large Stabilized Ship in Service, Including the Plant and Economies Effected by Stabilization," by Elmer A. Sperry, Esq., Member.

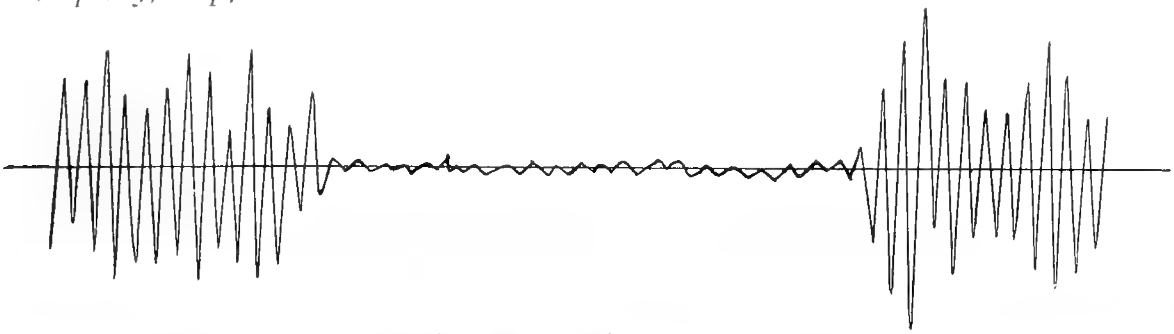

Fig. 4.—Characteristic Stabilization Curve.

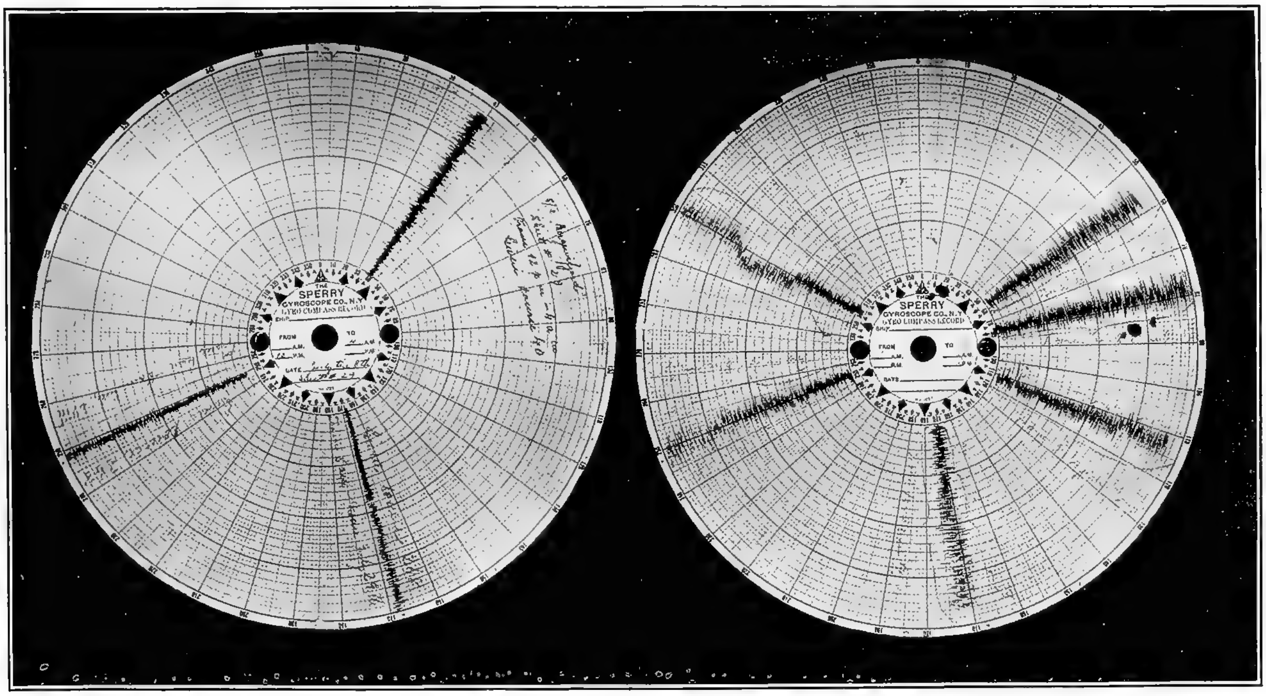

Fig. 5.—Automatic Yawing Records Showing at Right Increase Due to Rolling.

Fig. 6.—Automatic Record; Helm Required, Ship Not Rolling (Straight Line Indicates Rudder Center with One-Minute Intervals as Jogs.)

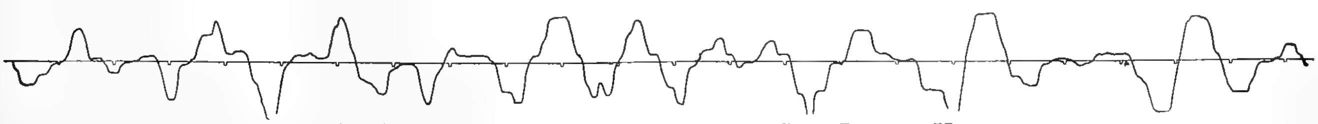

Fig. 7.—Automatic Record; Helm Required, Ship Rolling Heavily.

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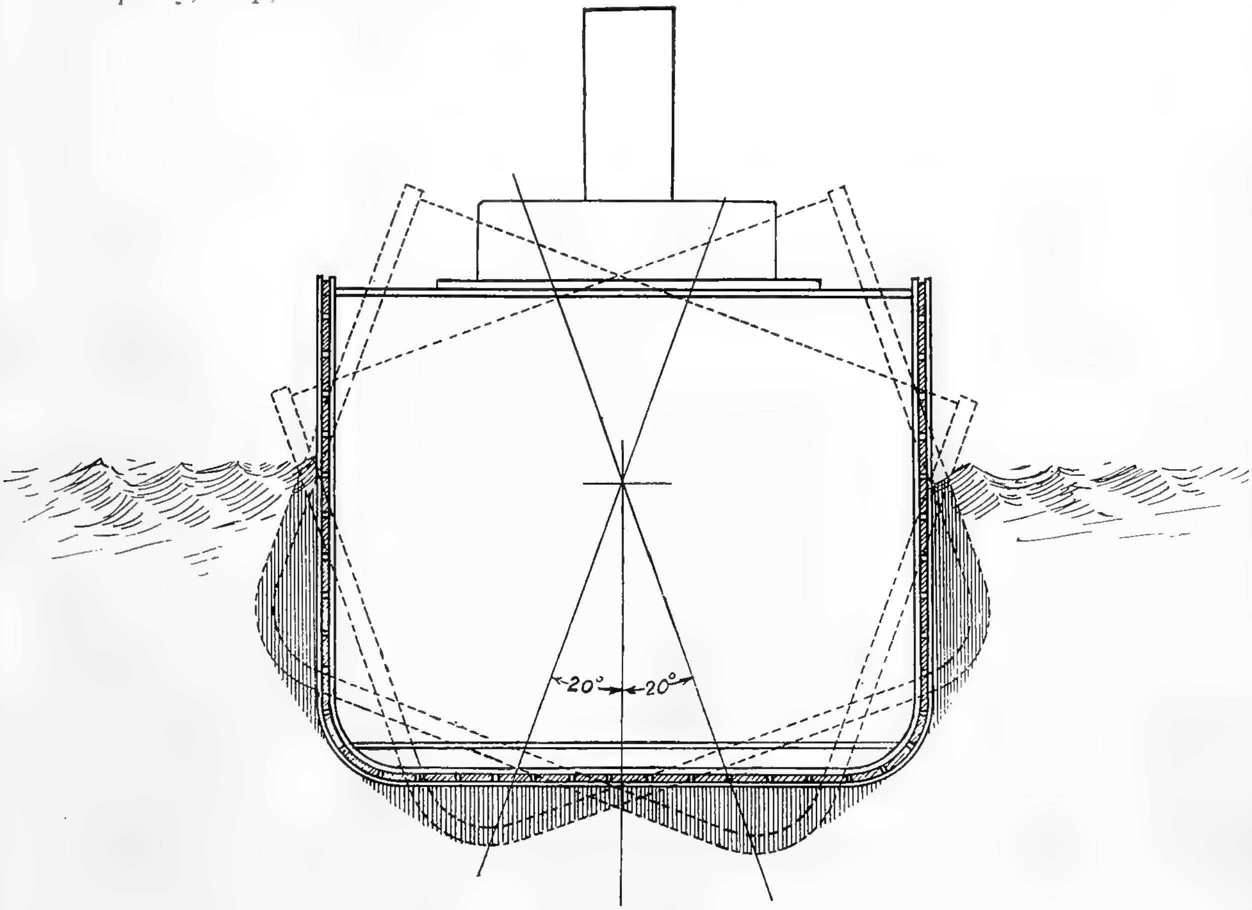

RELATION OF POWER CONSUMPTION TO ROLL.

The shaded area shows extra water disturbed. For a 37,500-ton vessel at 22 knots this means a loss of 3,100 horse-power.

FIG. 8.—POWER LOST BY ROLLING VESSEL.

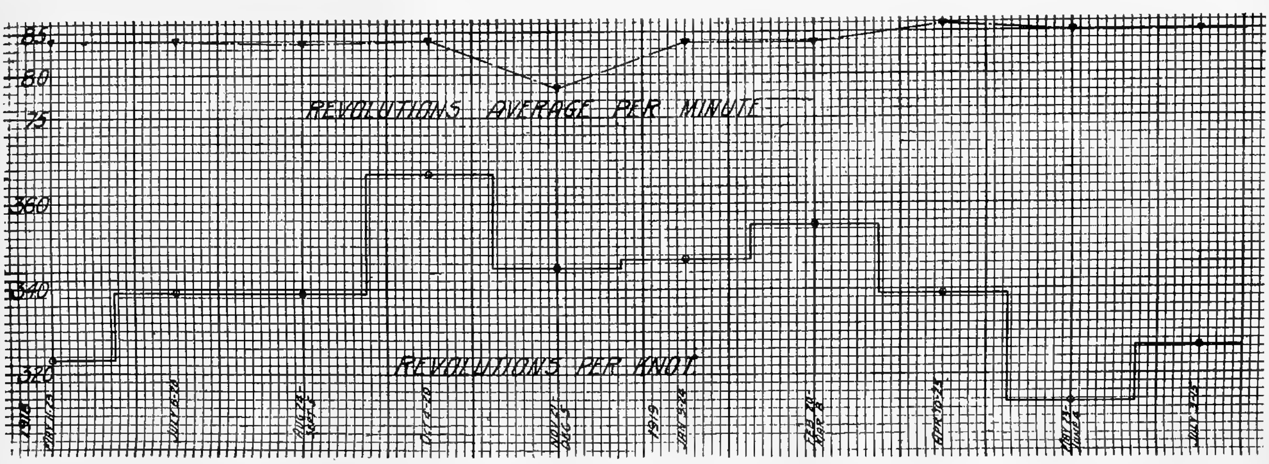

FIG. 9.—DATA FROM ENGINEER'S LOG OF CARGO AND PASSENGER VESSEL SHOWING RETARDATION BY WEATHER CONDITIONS IN WINTER MONTHS.

To illustrate paper on "Non-Rolling Passenger Liners—Observations on a Large Stabilized Ship in Service, Including the Plant and Economics Effected by Stabilization," by Elmer A. Sperry, Esq., Member.

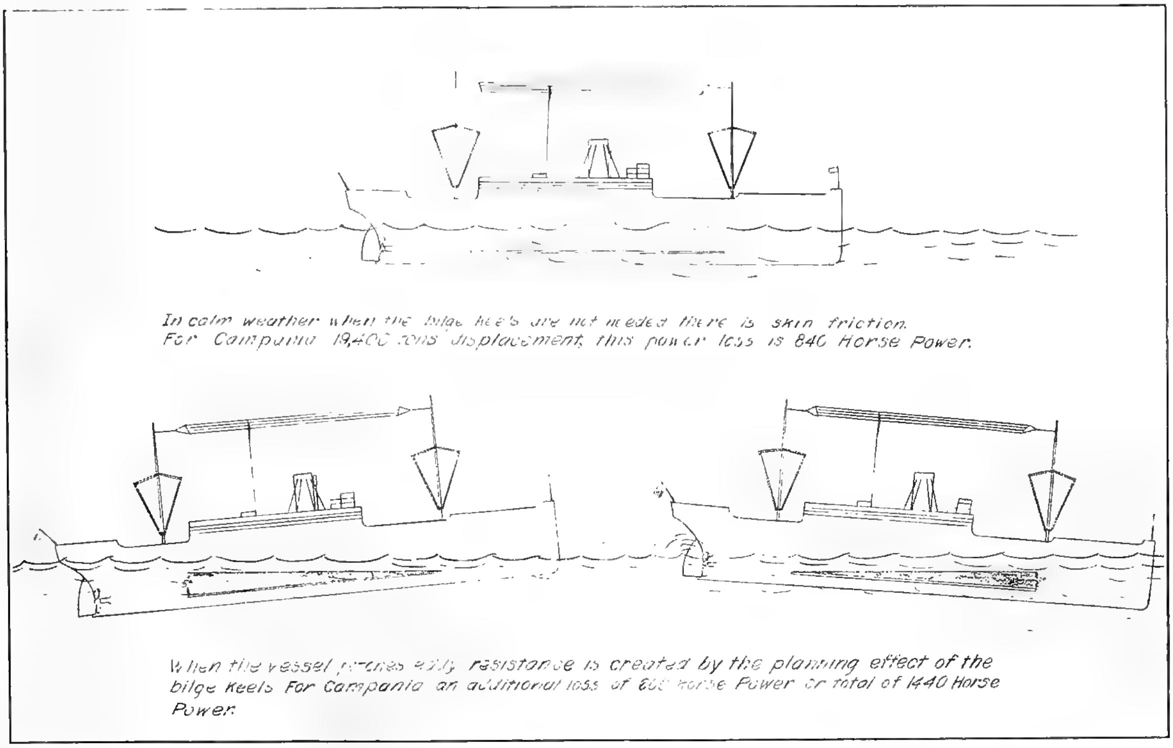

Fig. 10.—Power Absorption by Bilge Keels Which Are Omitted on Stabilized Vessels.

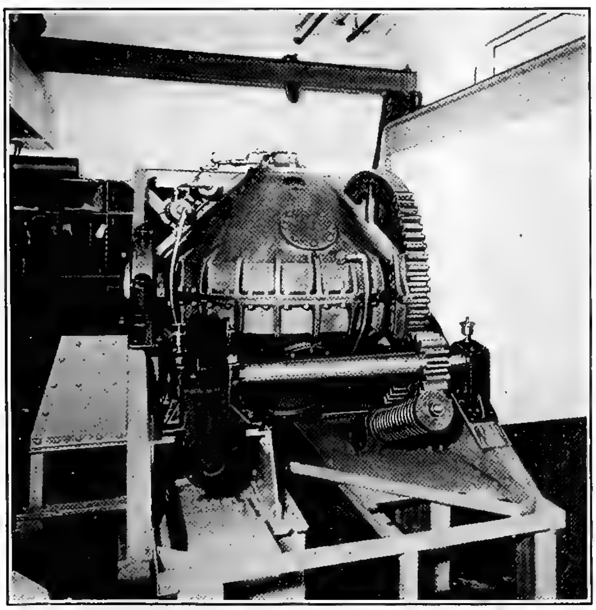

Fig. 11.—Complete Stabilizer Plant for 530-Ton Yacht Assembled in Shop for Test.

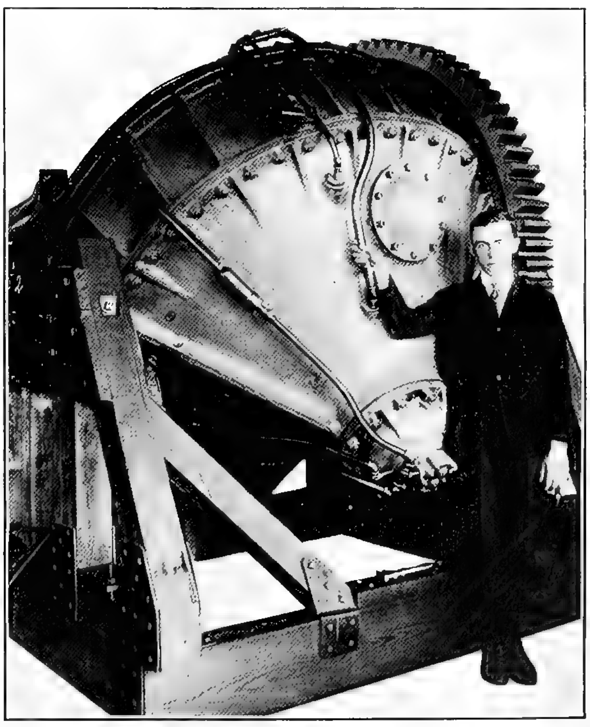

Fig. 12.—Stabilizer for Light Cruiser in Shipping Frame.

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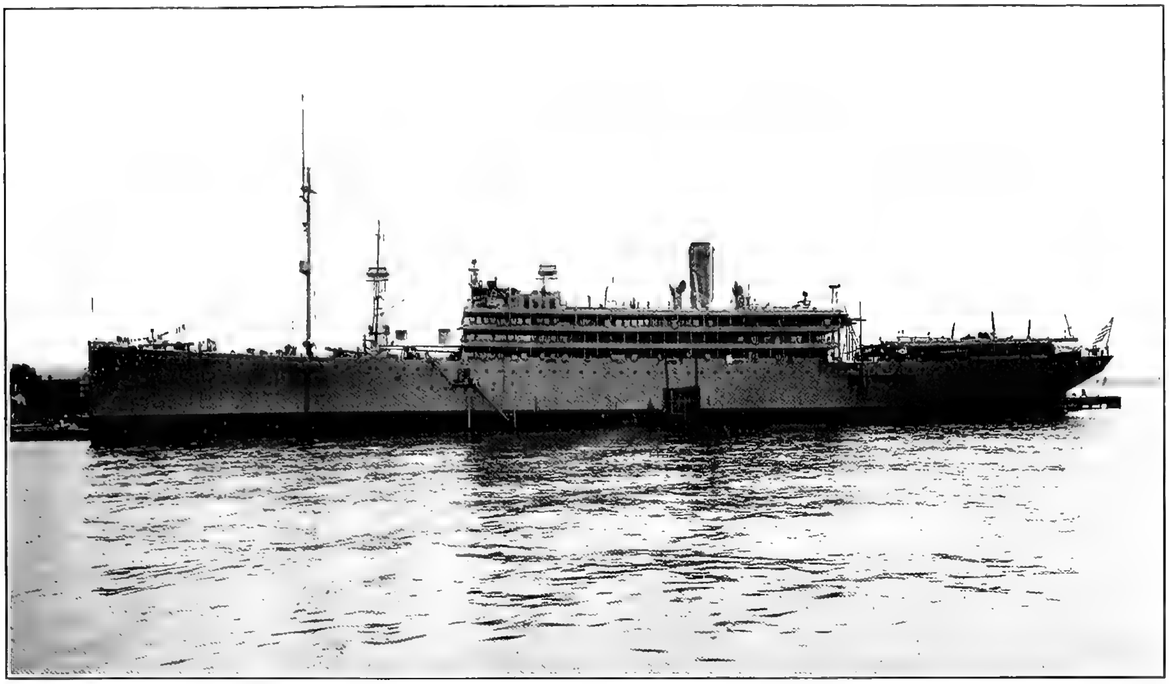


Fig. 13.—U. S. S. Henderson in Commission. Patch is Over Twice the Size of Stabilizer.

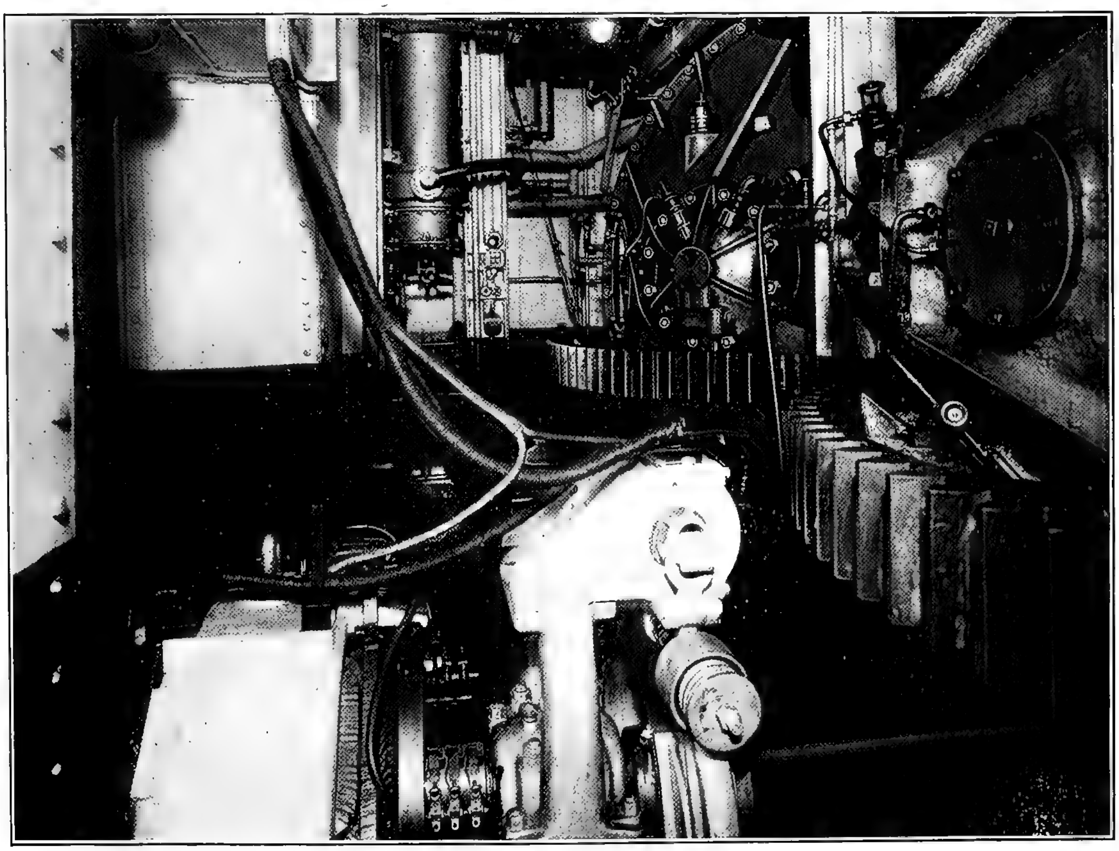


Fig. 14.—Stabilizer Room on U. S. S. Henderson. Main Gyros on Right, Precession Motor in Foreground.

To illustrate paper on "Non-Rolling Passenger Liners—Observations on a Large Stabilized Ship in Service, Including the Plant and Economics Effected by Stabilization," by Elmer A. Sperry, Esq., Member.

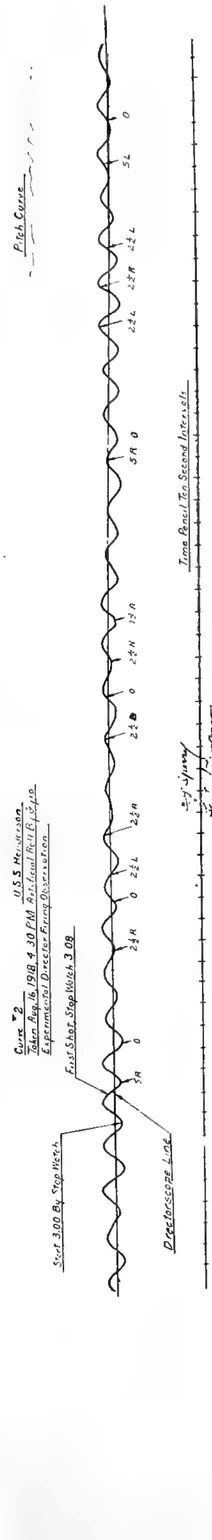


FIG. 15.—AUTOMATIC CONSTANT ROLL IMPARTED TO U. S. S. HENDERSON AT SEA.

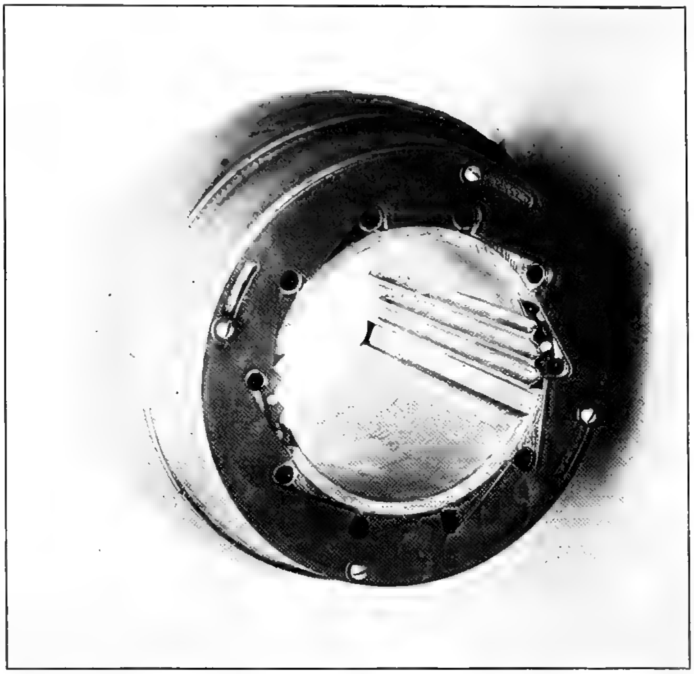


FIG. 17.—MAIN GYRO BEARING PHOTOGRAPHED AFTER 10,000 PRECESSION LOADS. TOOL MARKS STILL VISIBLE.

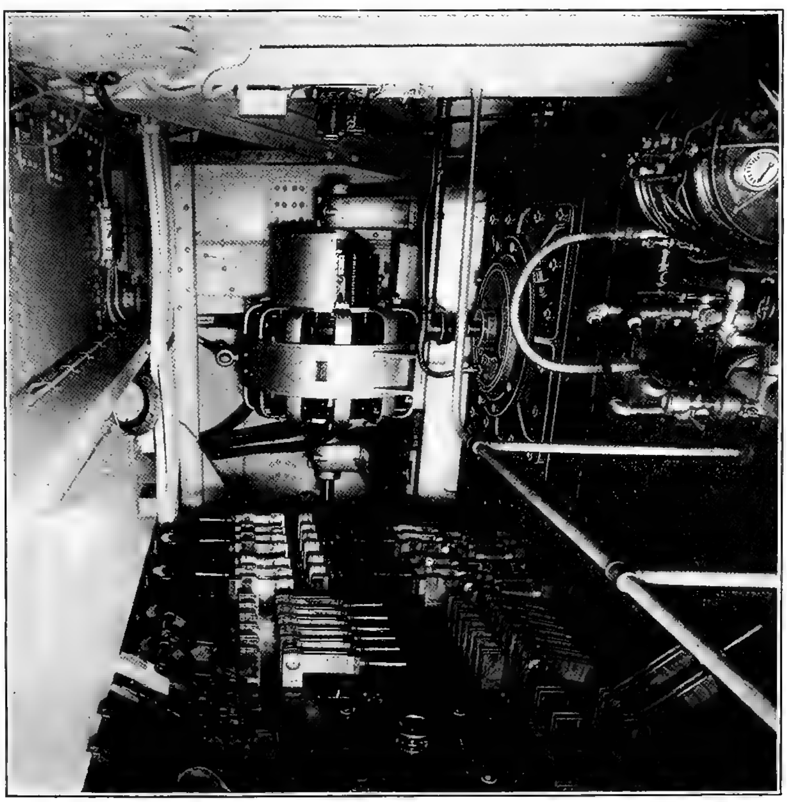
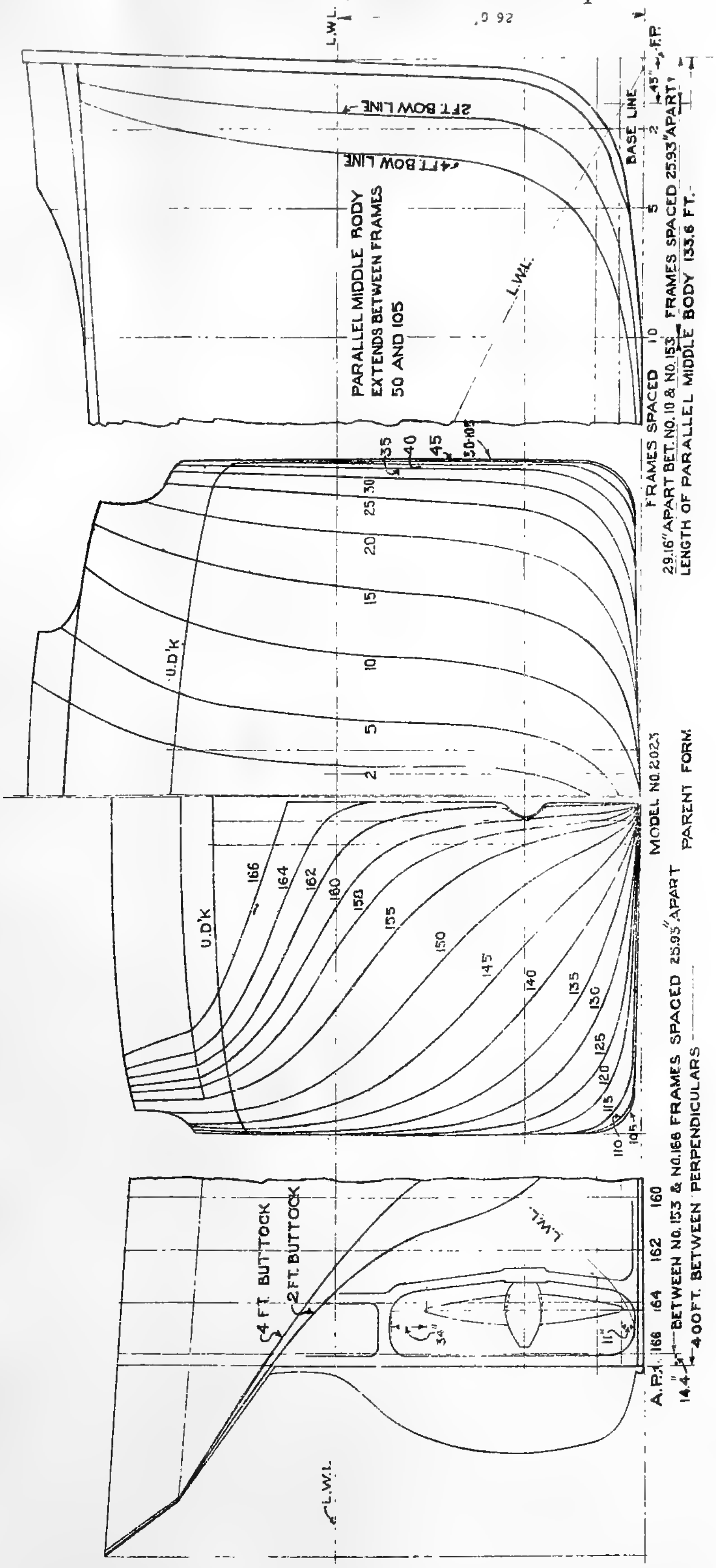


FIG. 16.—CONTROL ROOM, U. S. S. HENDERSON STABILIZER.

To illustrate paper on "The Propulsive Efficiency of Single Screw Cargo Ships." by Commander William McEntee, Construction Corps. U. S. N., Member.



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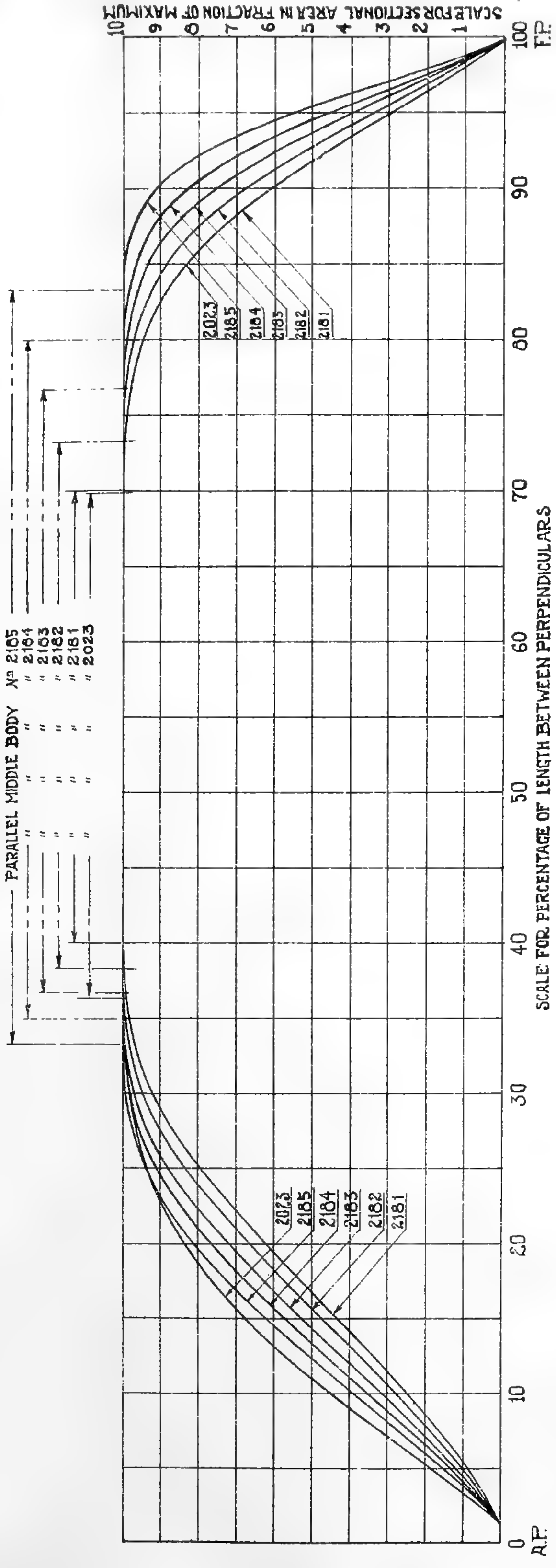

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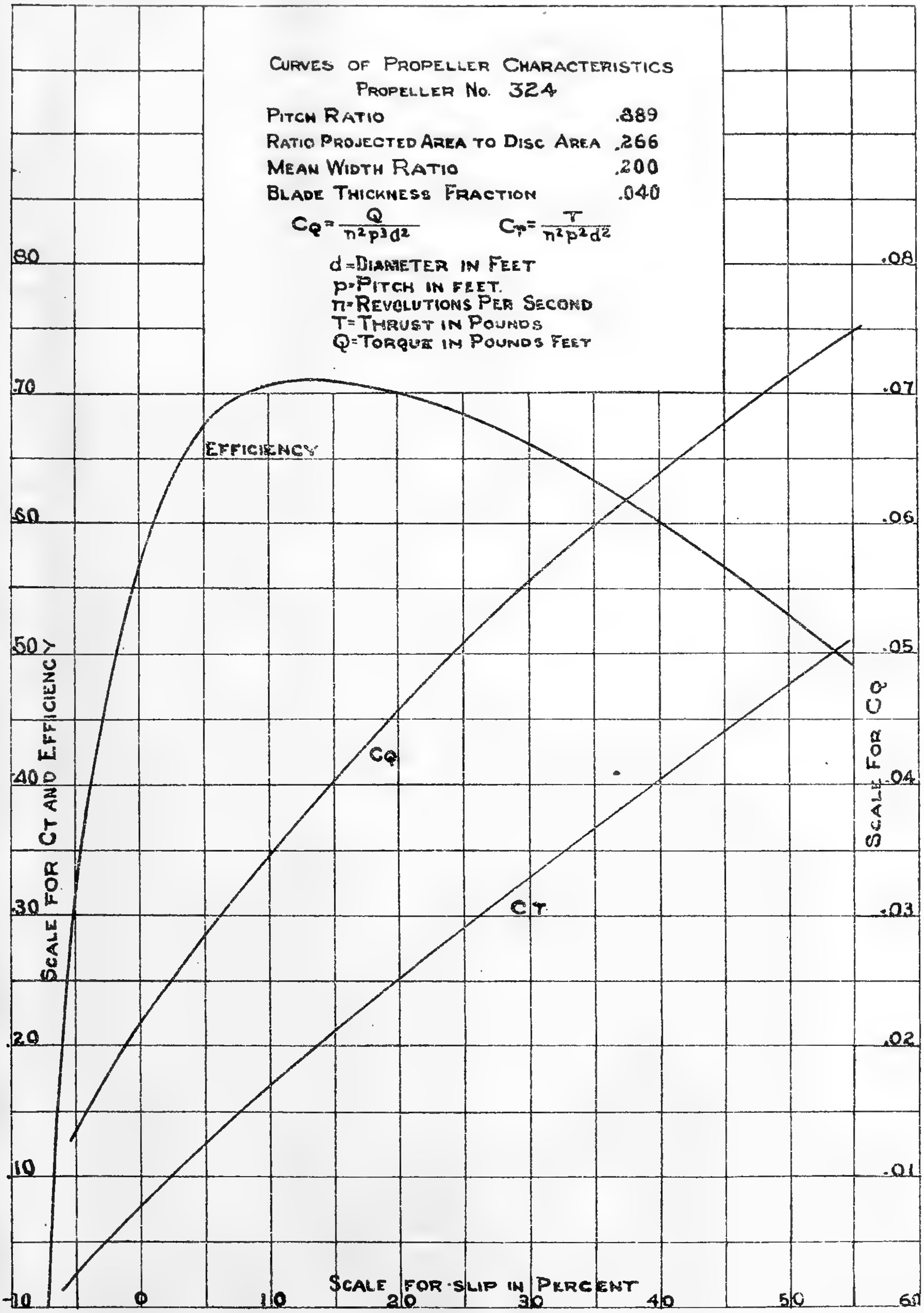

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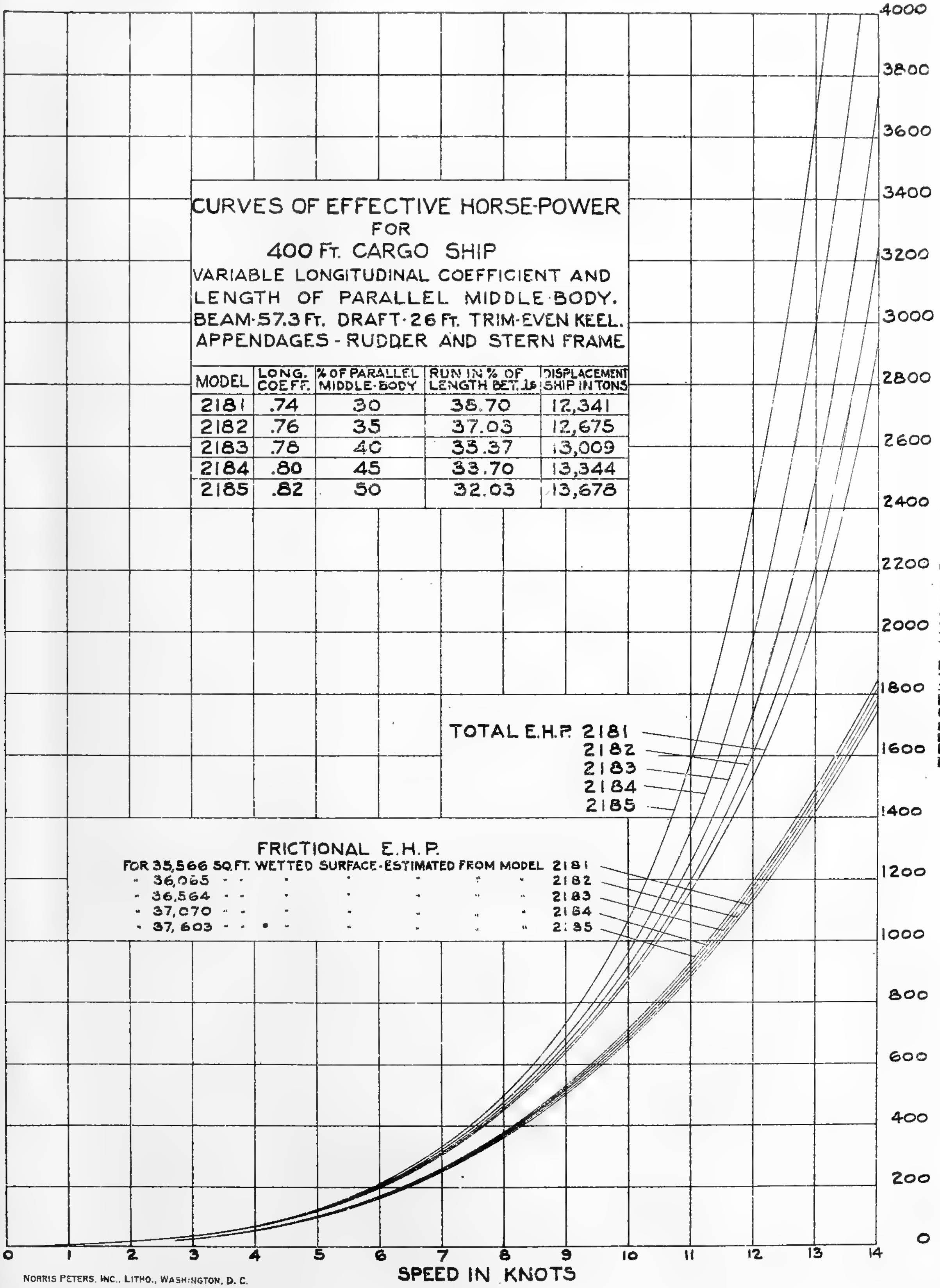

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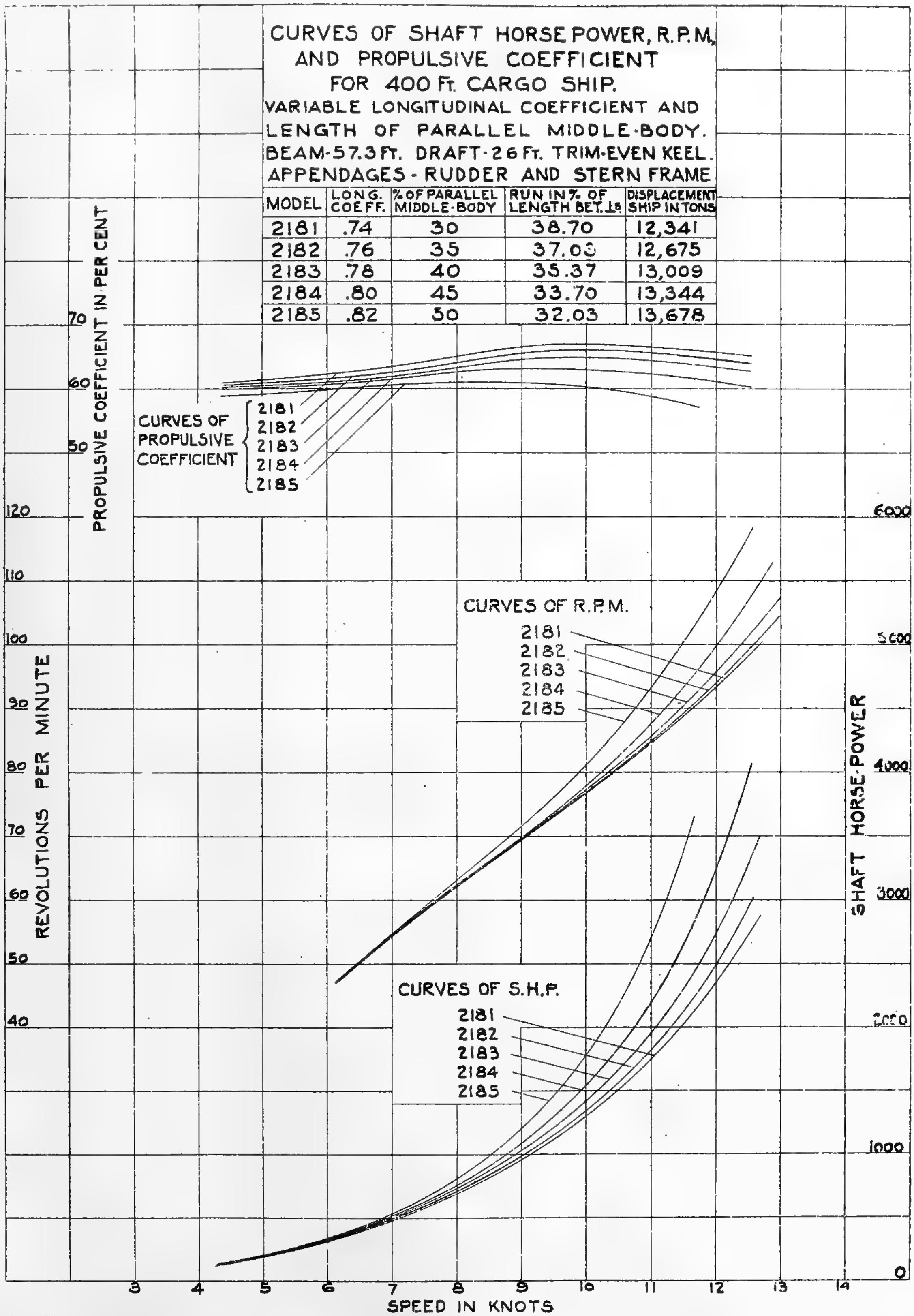

MODEL	LONG. COEFF.	% OF PARALLEL MIDDLE-BODY	RUN IN % OF LENGTH BET. ⊥s	DISPLACEMENT SHIP IN TONS
2181	.74	30	38.70	12,341
2182	.76	35	37.03	12,675
2183	.78	40	35.37	13,009
2184	.80	45	33.70	13,344
2185	.82	50	32.03	13,678

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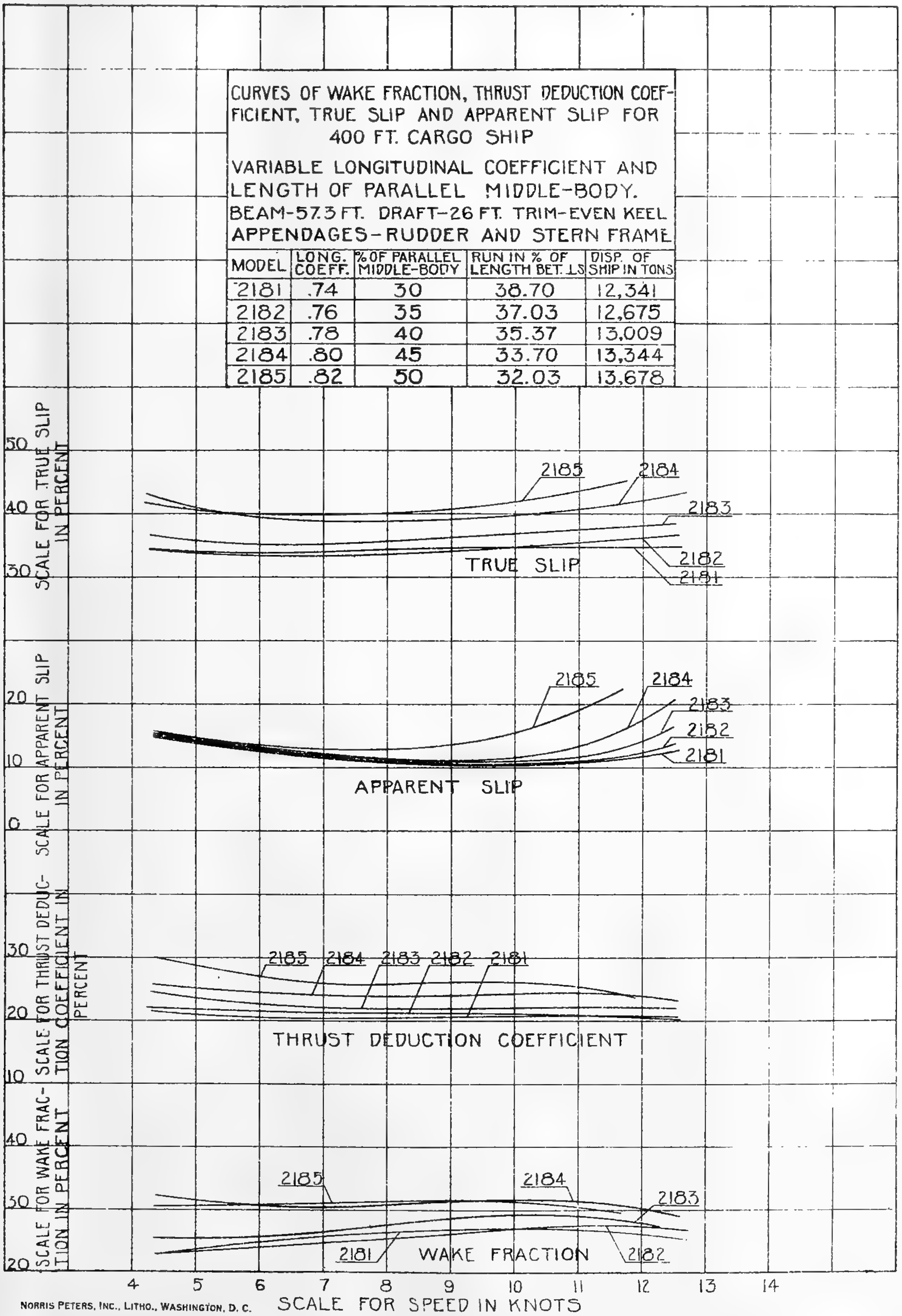

MODEL	LONG. COEFF.	% OF PARALLEL MIDDLE-BODY	RUN IN % OF LENGTH BET. ⊥S	DISP. OF SHIP IN TONS
2181	.74	30	38.70	12,341
2182	.76	35	37.03	12,675
2183	.78	40	35.37	13,009
2184	.80	45	33.70	13,344
2185	.82	50	32.03	13,678

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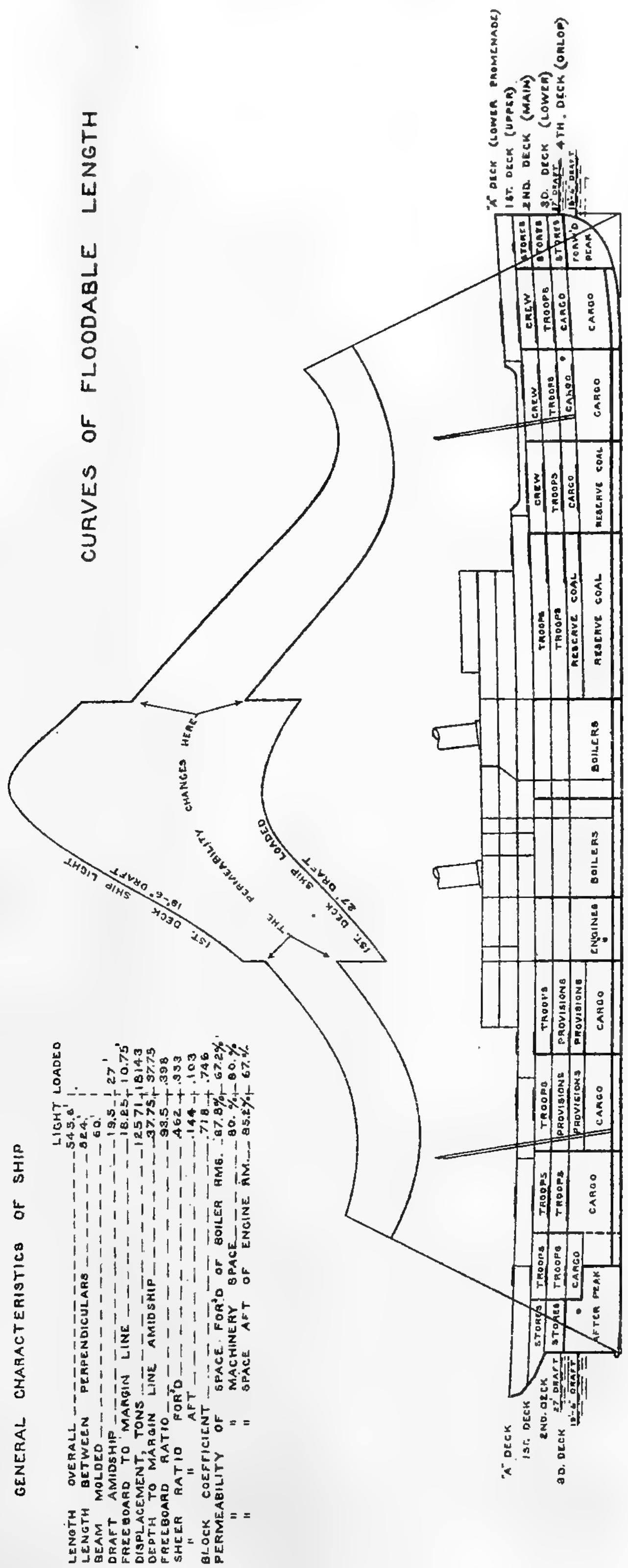
To illustrate paper on "Buoyancy and Stability of Troop Transports," by Professor William Hovgaard, Member.



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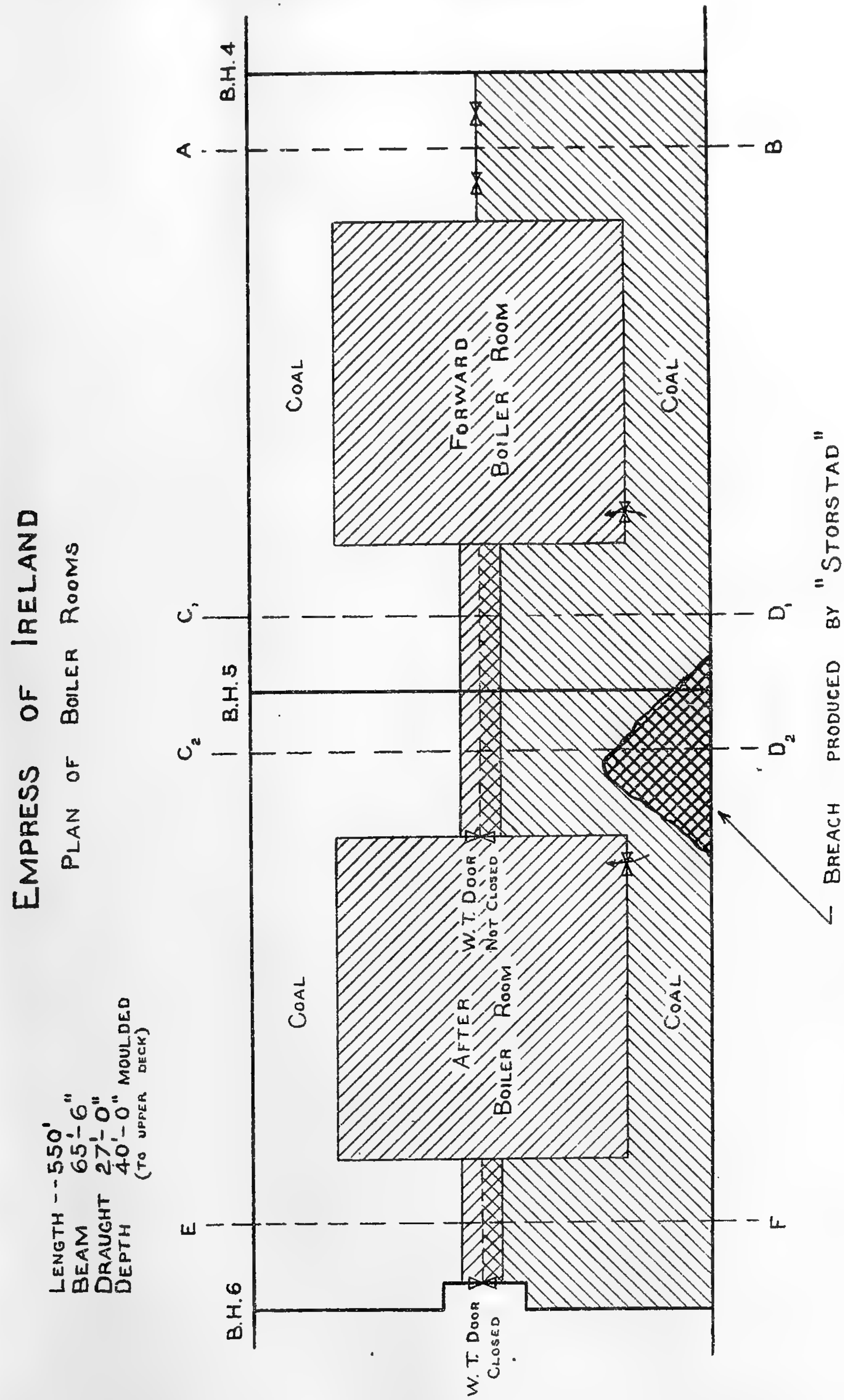

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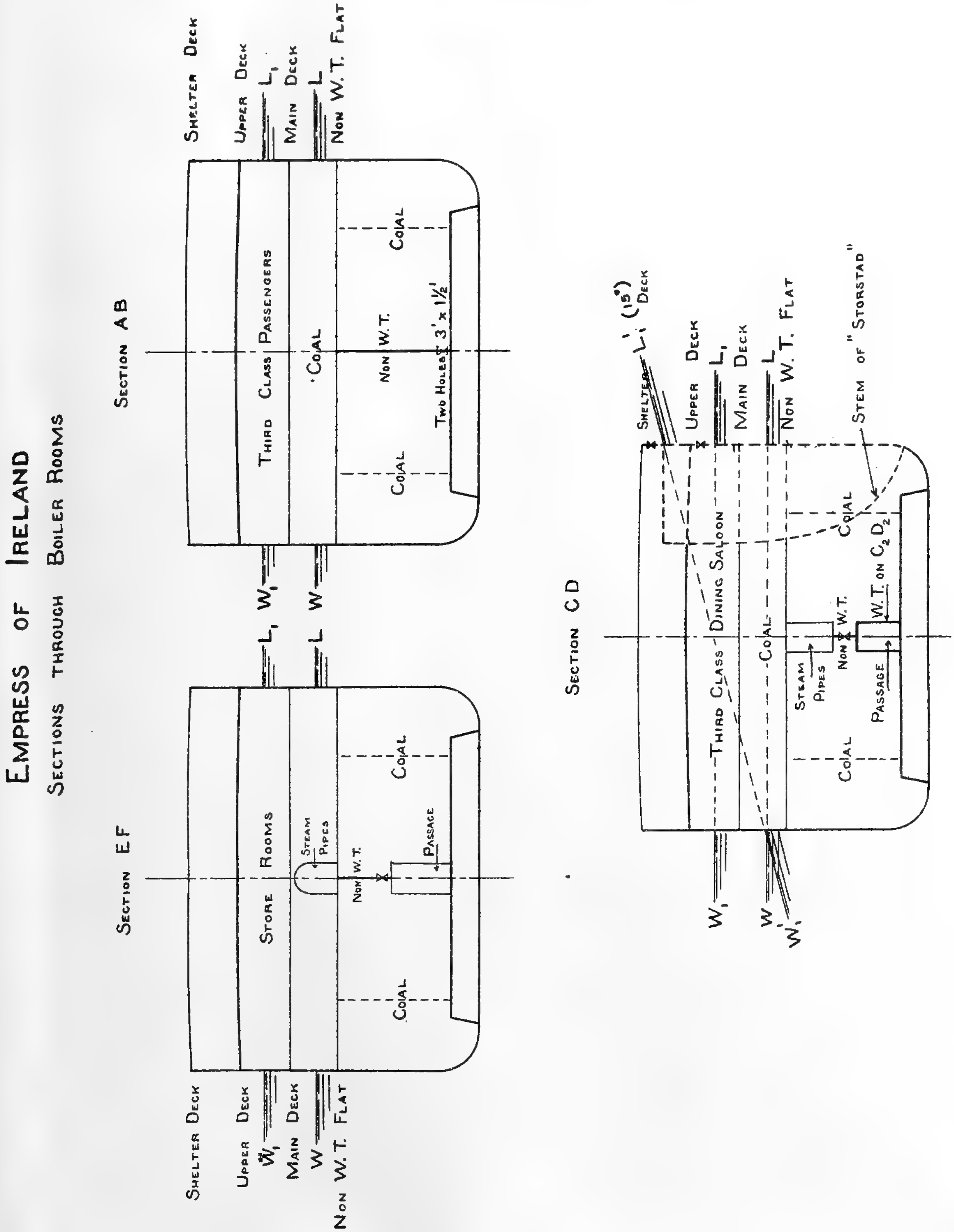

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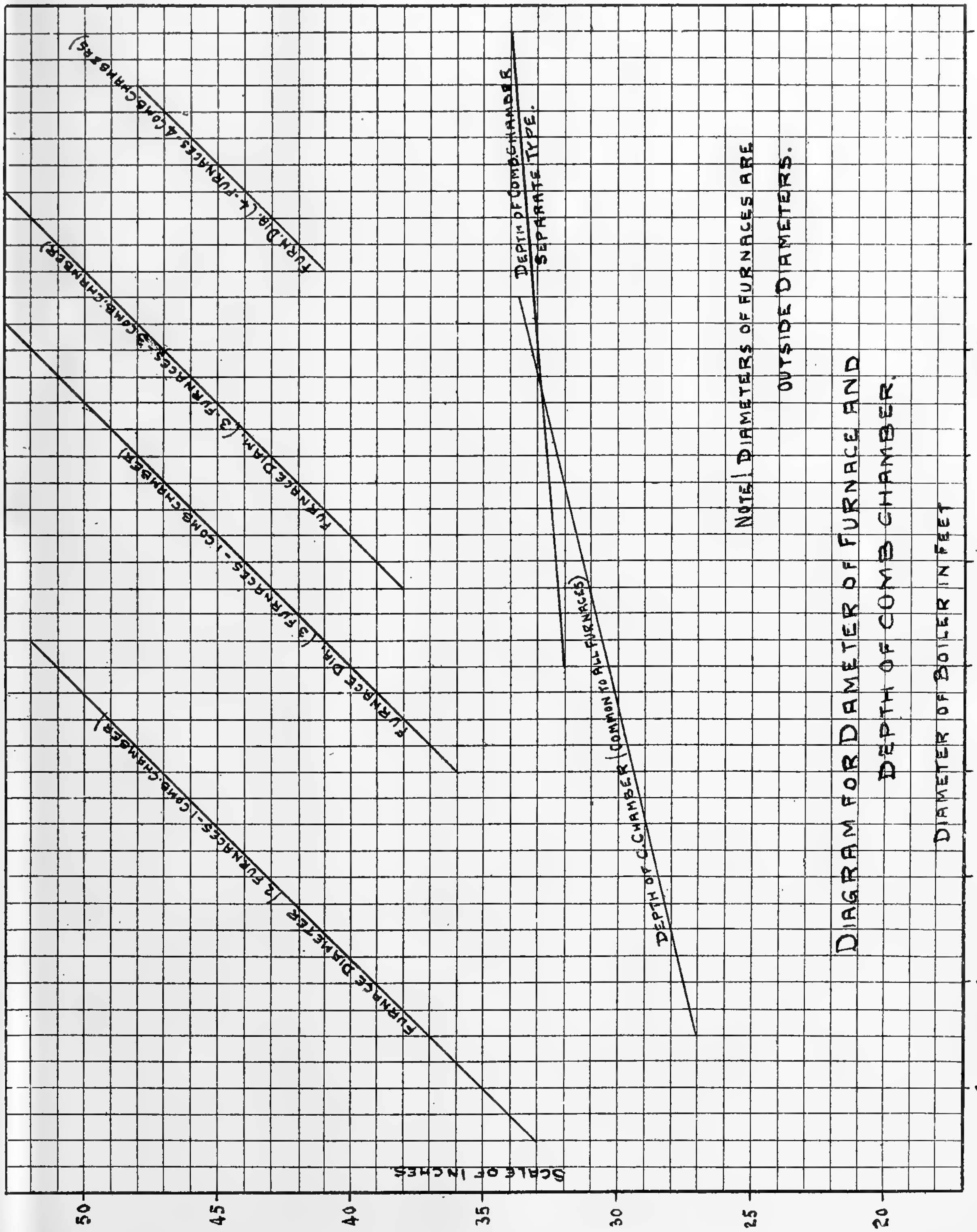
To illustrate paper on "The Application of Standardization and Graphical Methods to the Design of Cylindrical Return Tubular Boilers," by Henry C. E. Meyer, Esq., Member.



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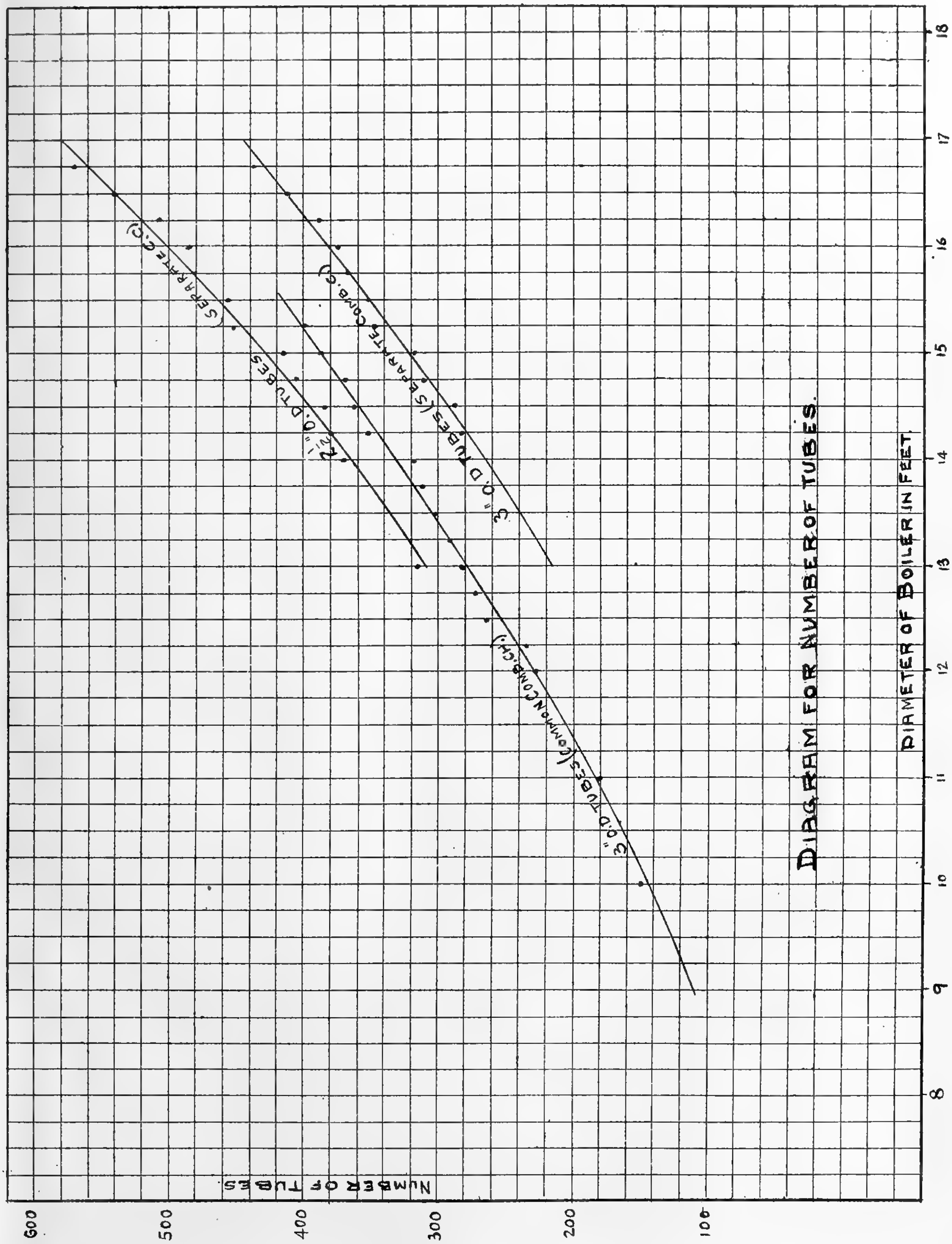

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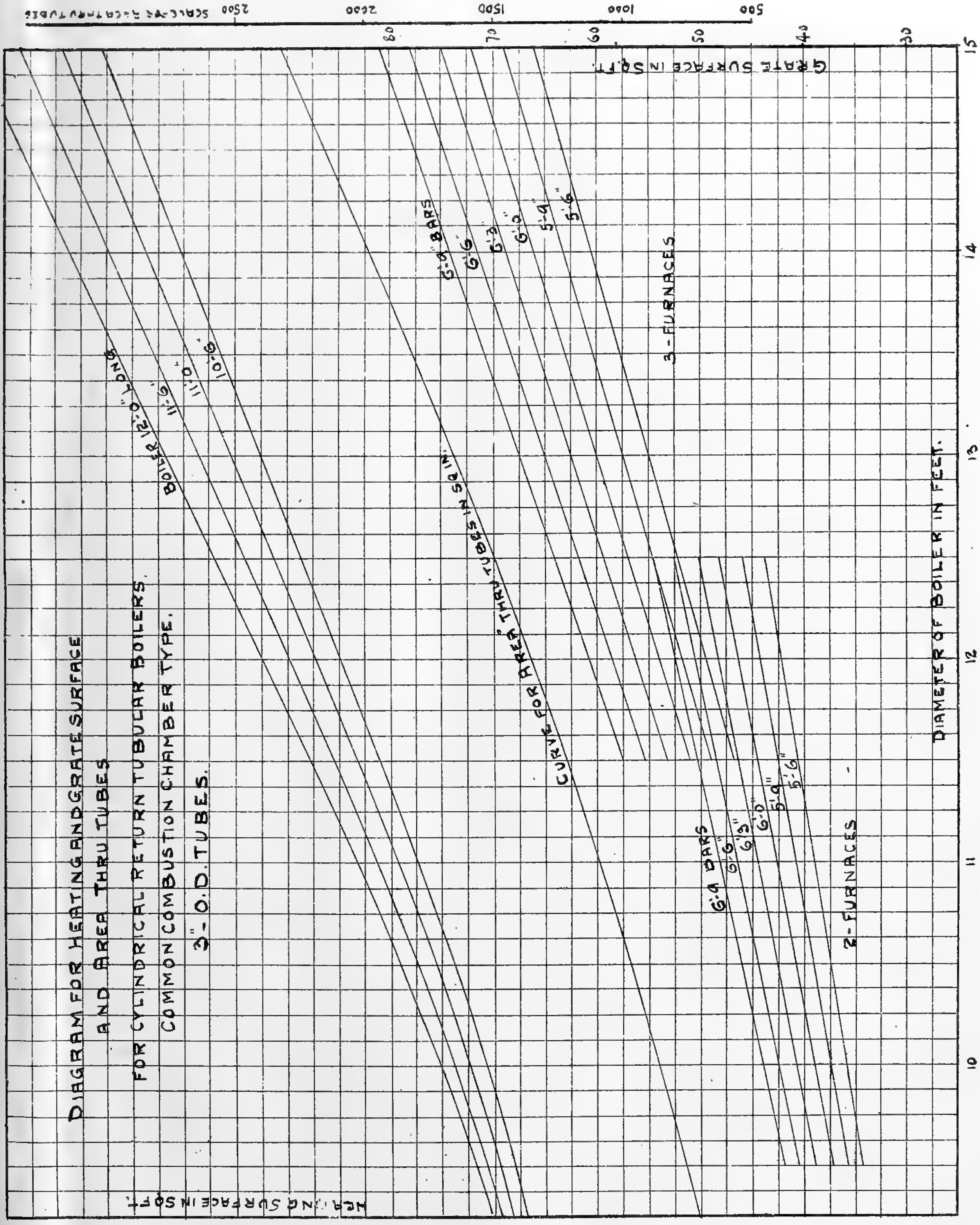

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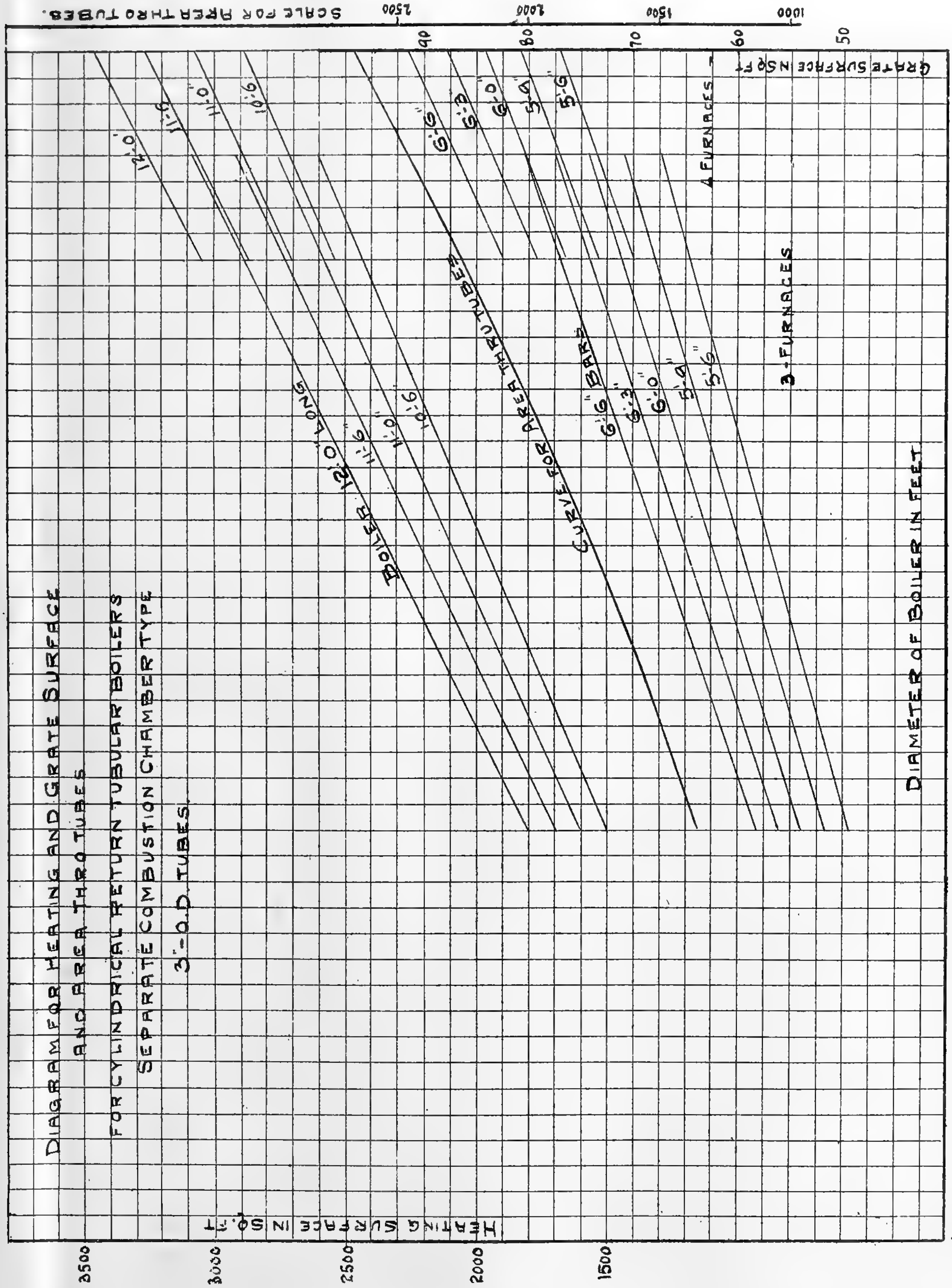

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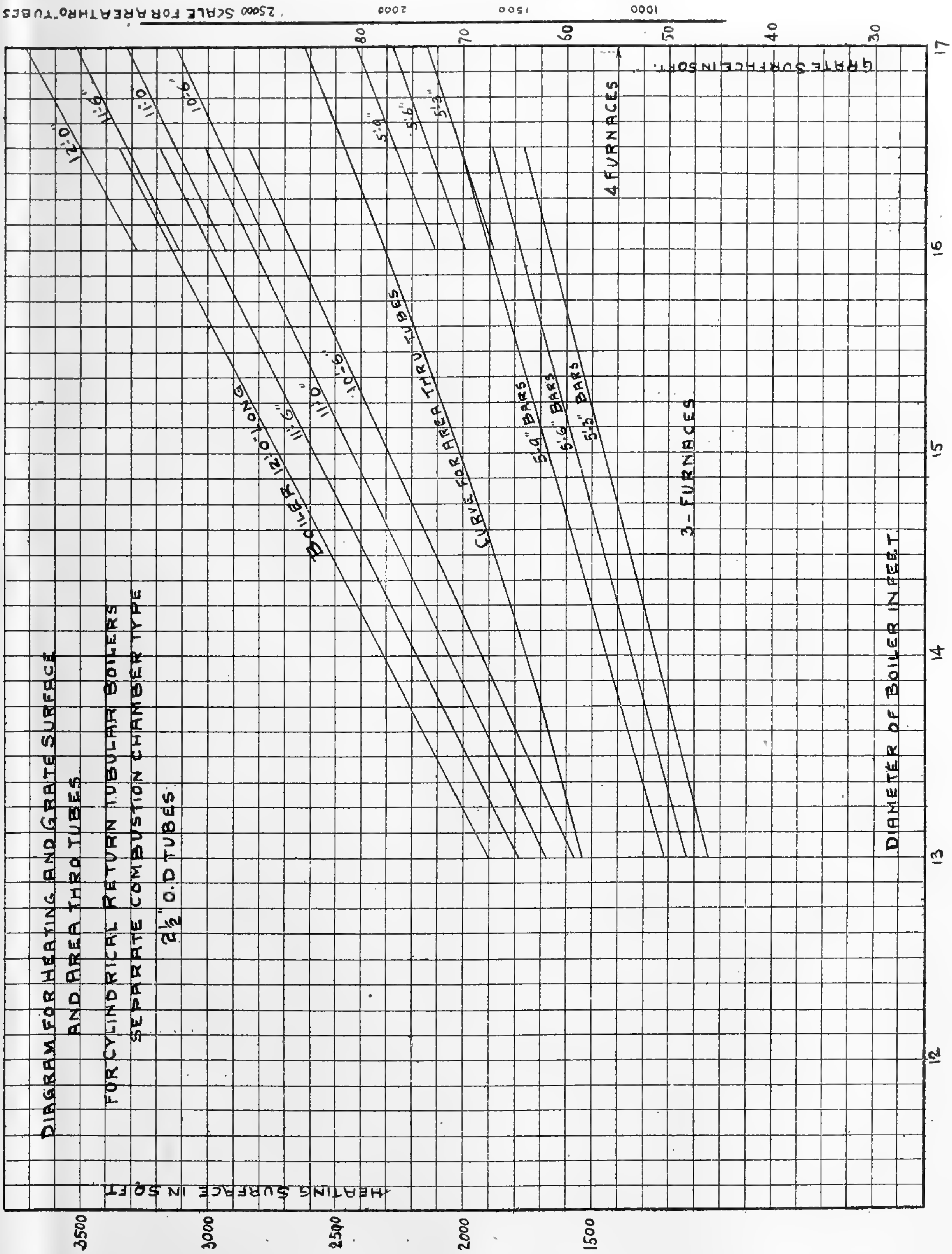

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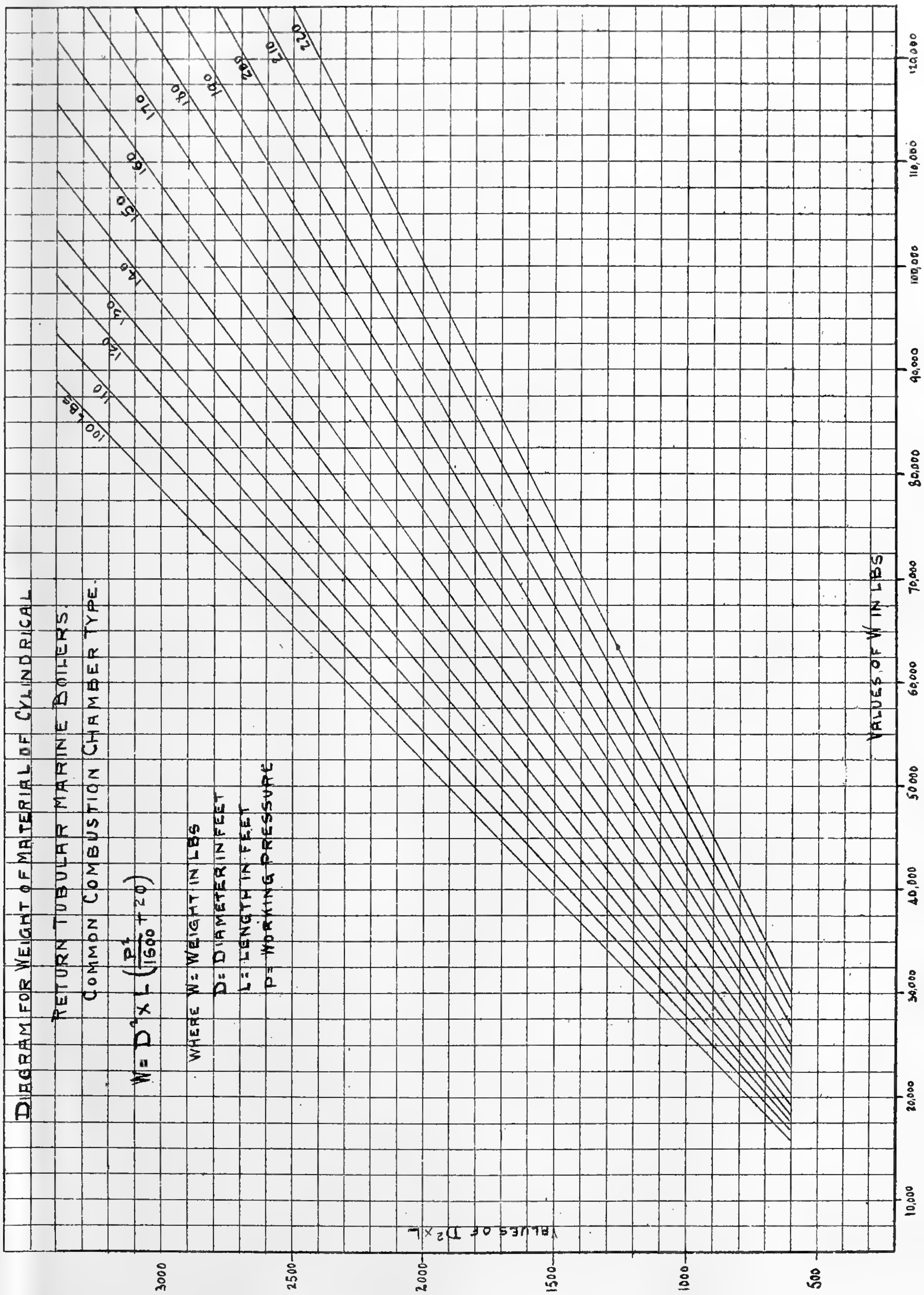

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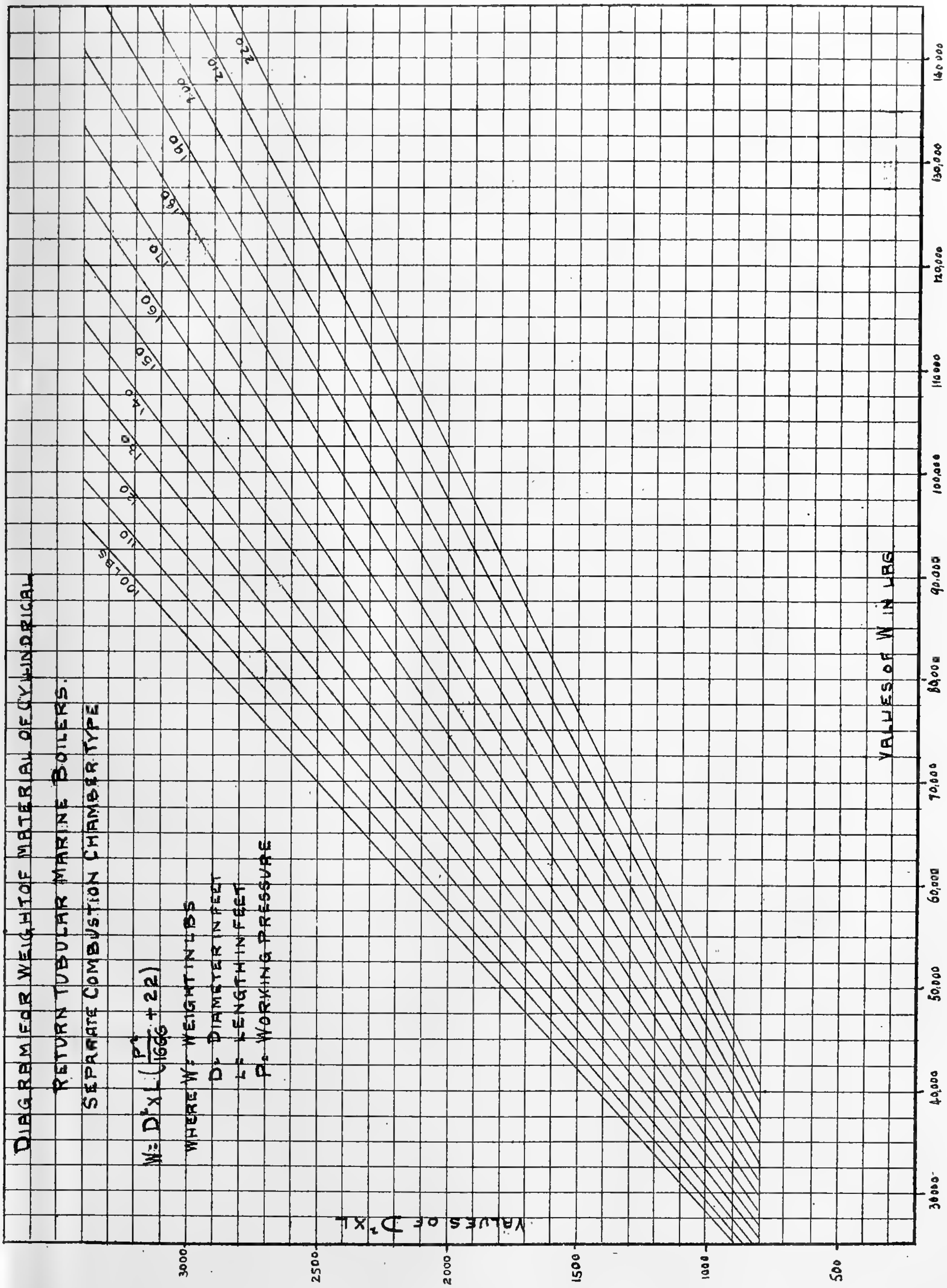

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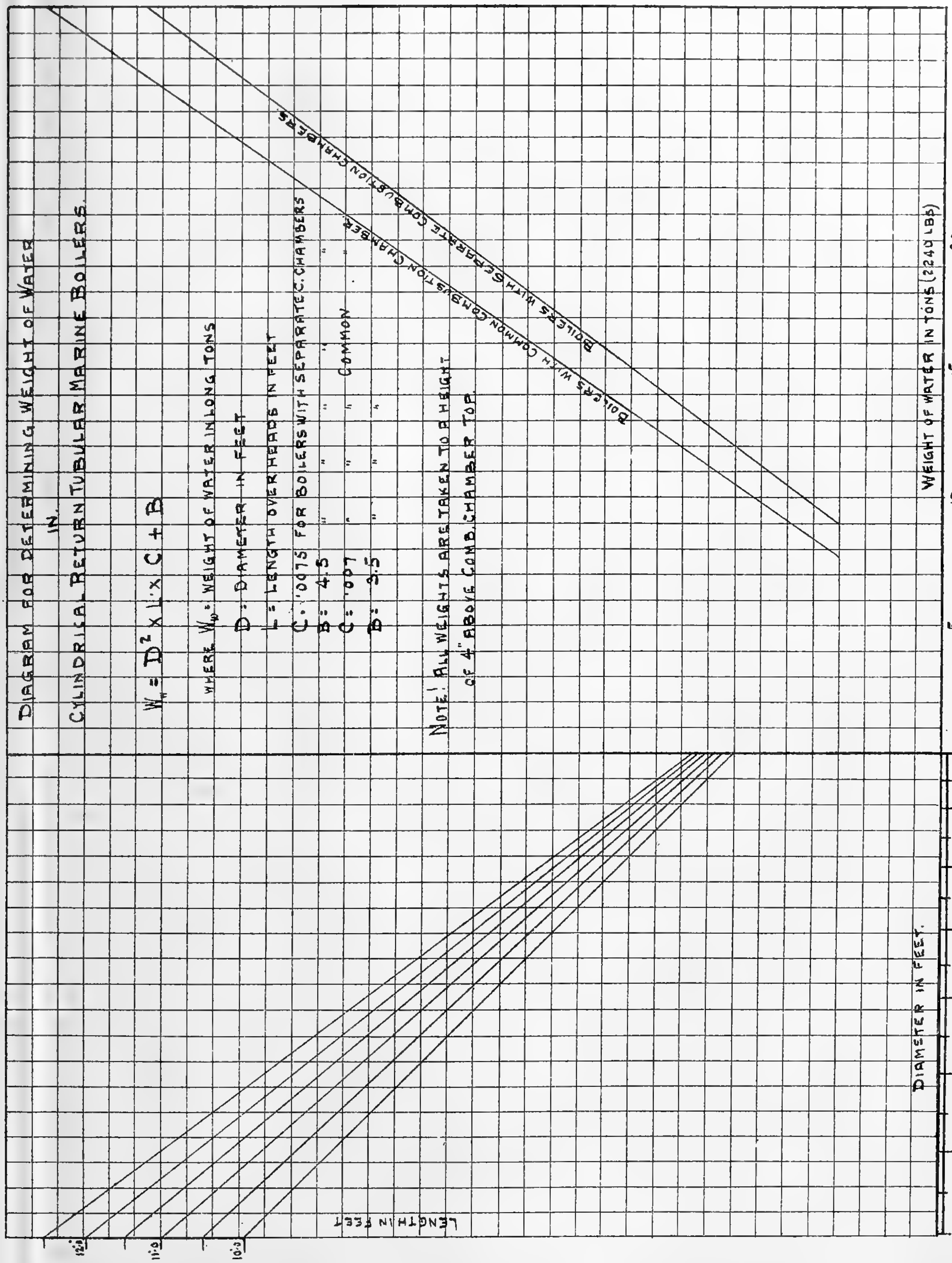

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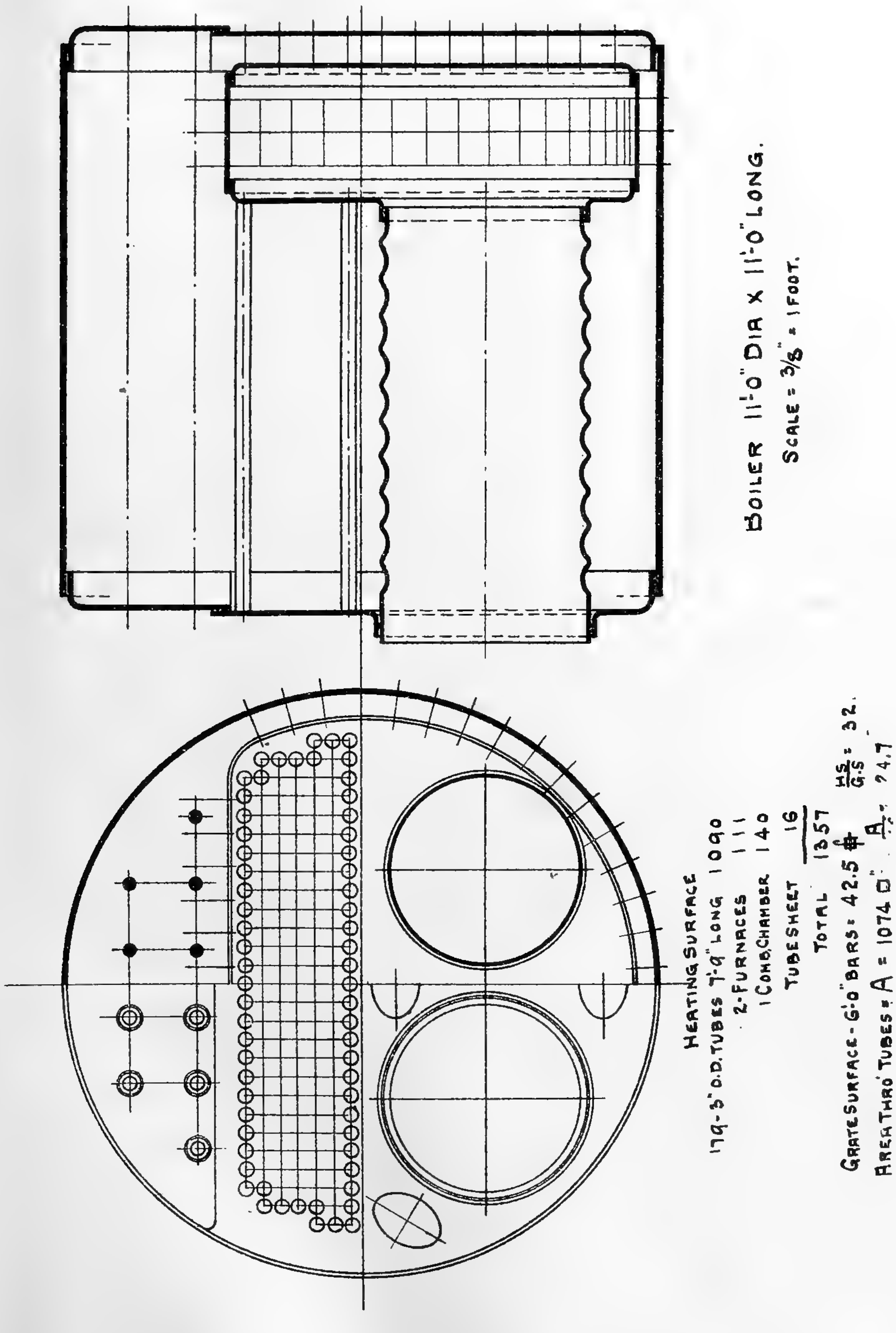
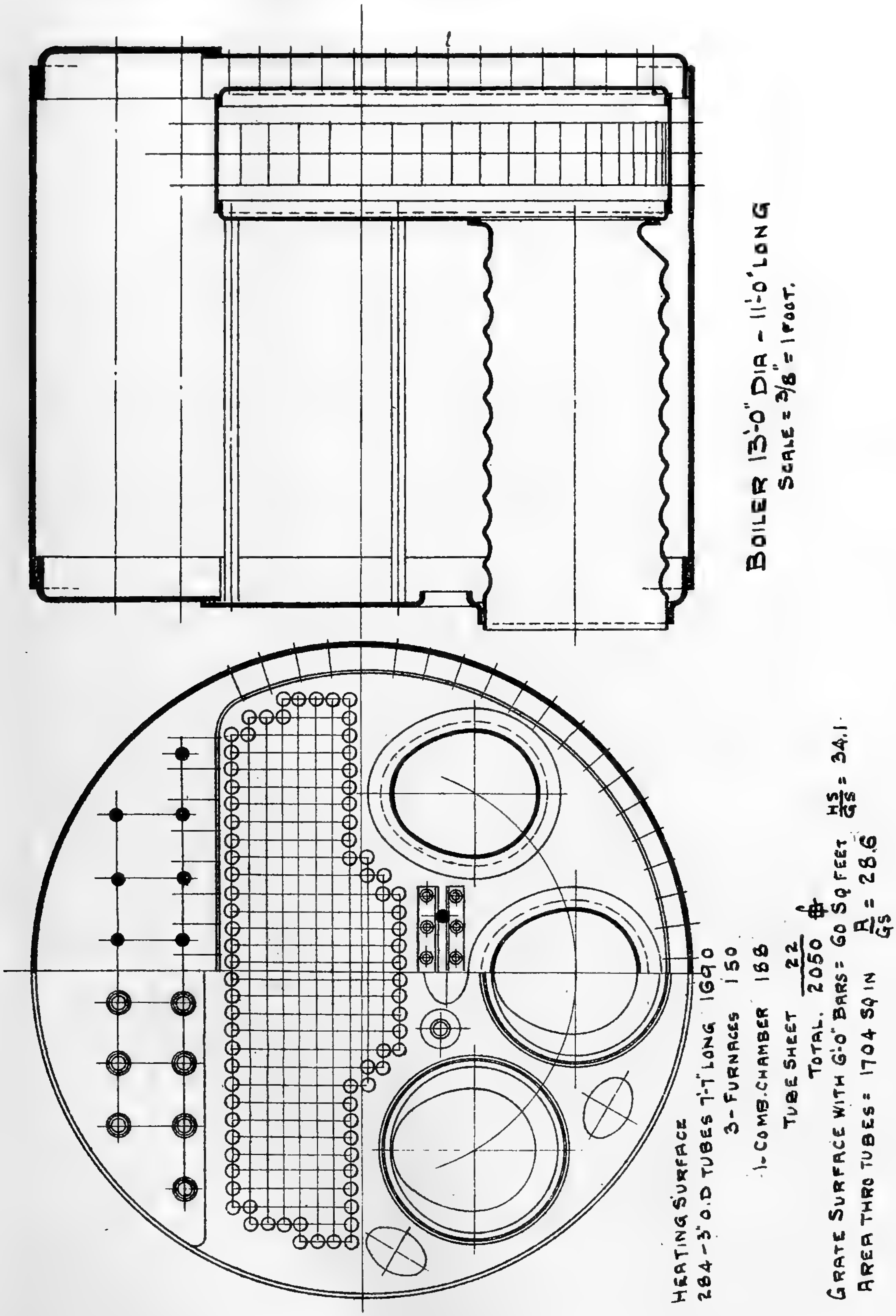
To illustrate paper on "The Application of Standardization and Graphical Methods to the Design of Cylindrical Return Tubular Boilers," by Henry C. E. Meyer, Esq., Member.



[illegible] re paper on "The Application of [illegible] and Graphical Methods to the [illegible] of Cylindrical Marine Tubular Boilers" by Henry C. H. [illegible], Esq. [illegible]

To illustrate paper on "The Application of Standardization and Graphical Methods to the Design of Cylindrical Return Tubular Boilers," by Henry C. E. Meyer, Esq., Member.



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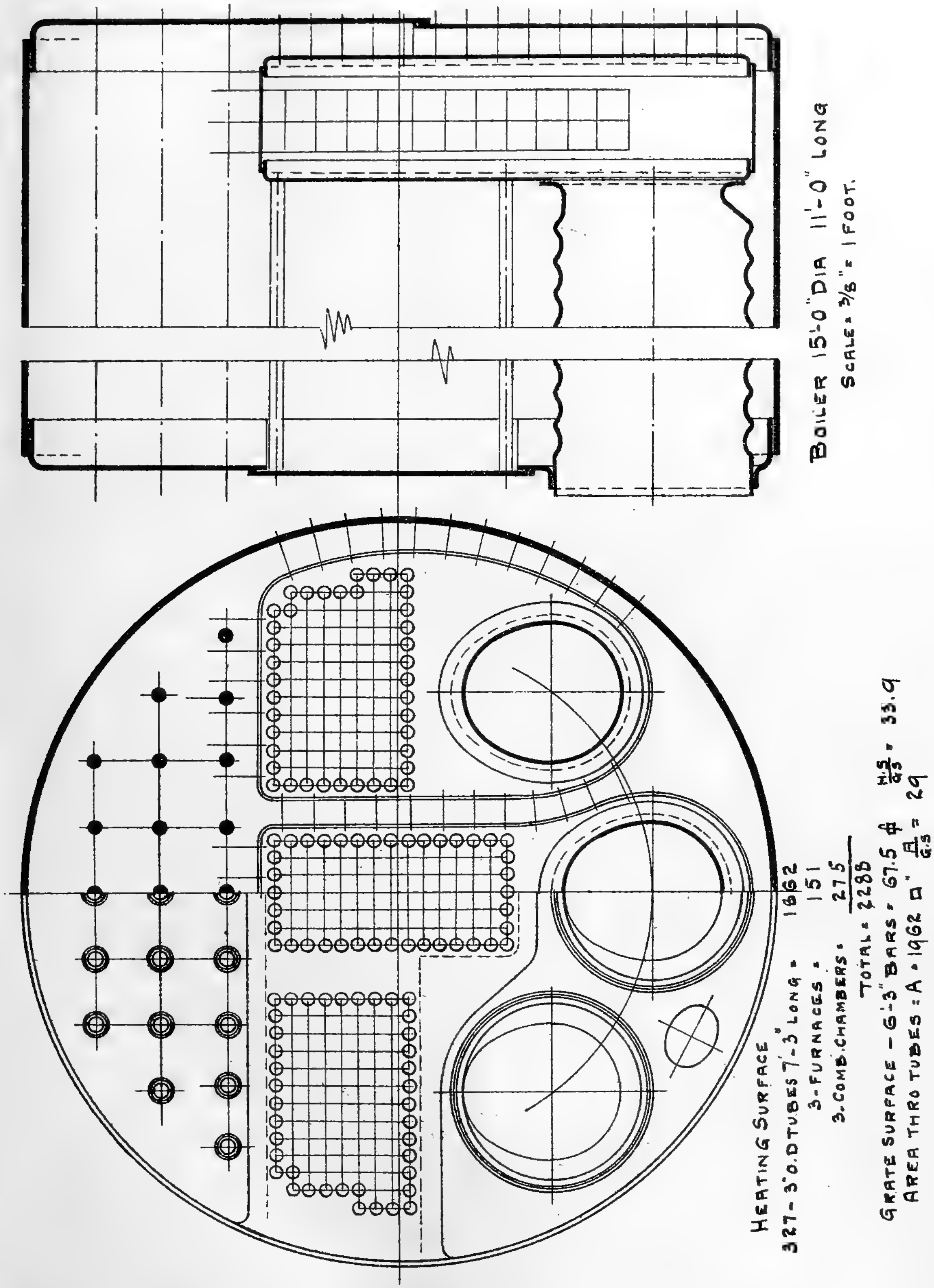
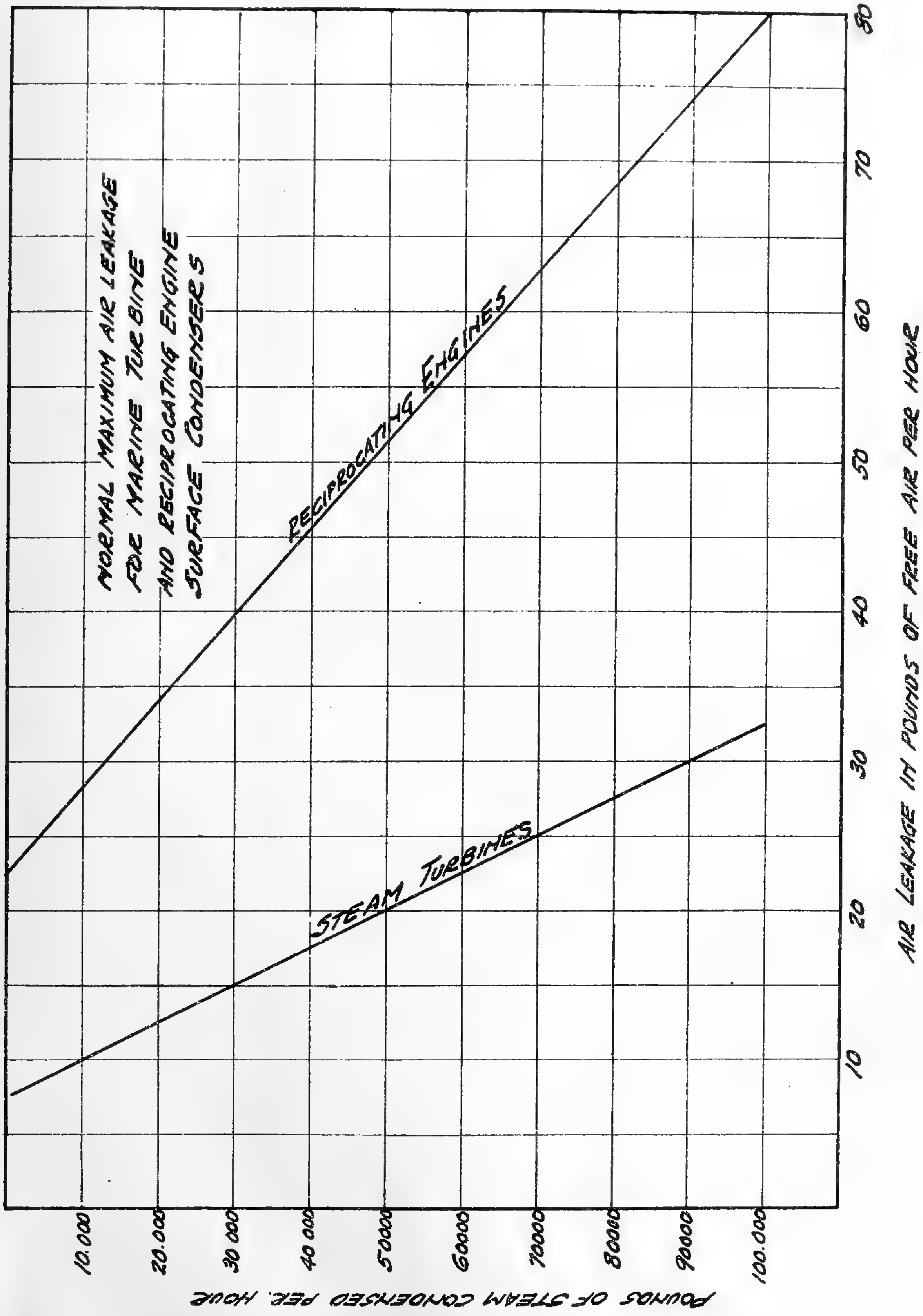
To illustrate paper on "The Application of Standardization and Graphical Methods to the Design of Cylindrical Return Tubular Boilers," by Henry C. E. Meyer, Esq., Member.



NORRIS PETERS, INC., LITHO., WASHINGTON, D. C.

To illustrate paper on "New Developments in High Vacuum Apparatus," by G. L. E. Kothny, Esq., Member.



Curves showing permissible air leakage for marine surface condensers.

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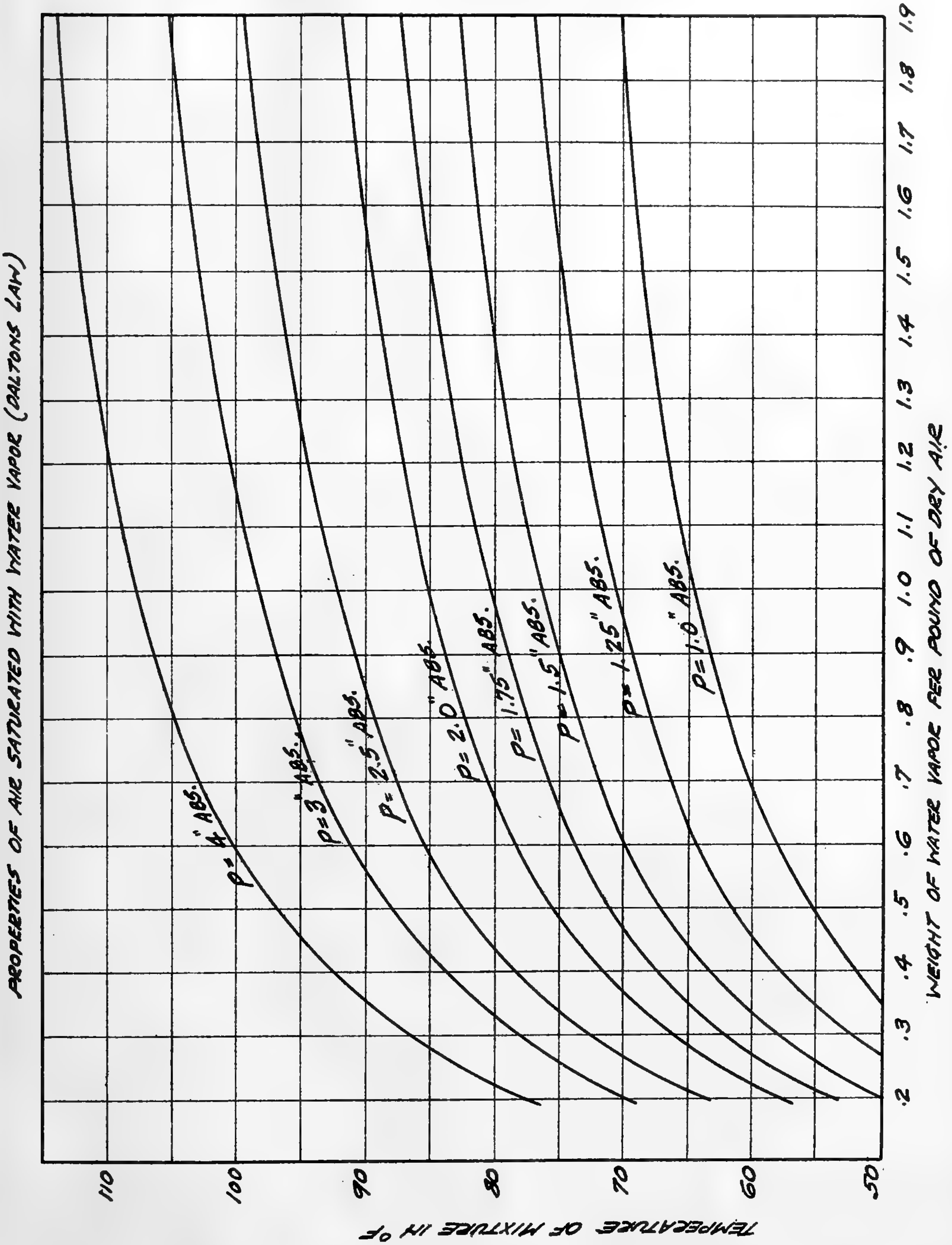

Curves giving weight of water vapor in pounds per pound of dry air for different absolute pressures and different temperatures.

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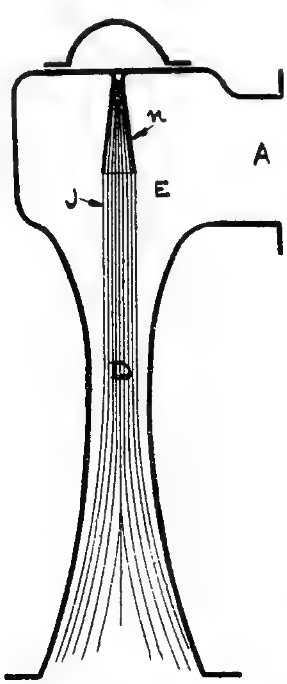
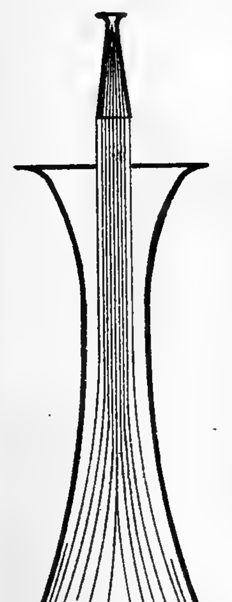
FIG. 3



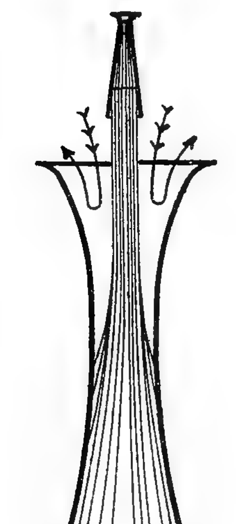
Diagramatic sketch of single stage ejector.

FIG. 4



Sketch showing diagramatically the proper working ejector.

FIG. 5



Sketch showing diagramatically the improper working ejector.

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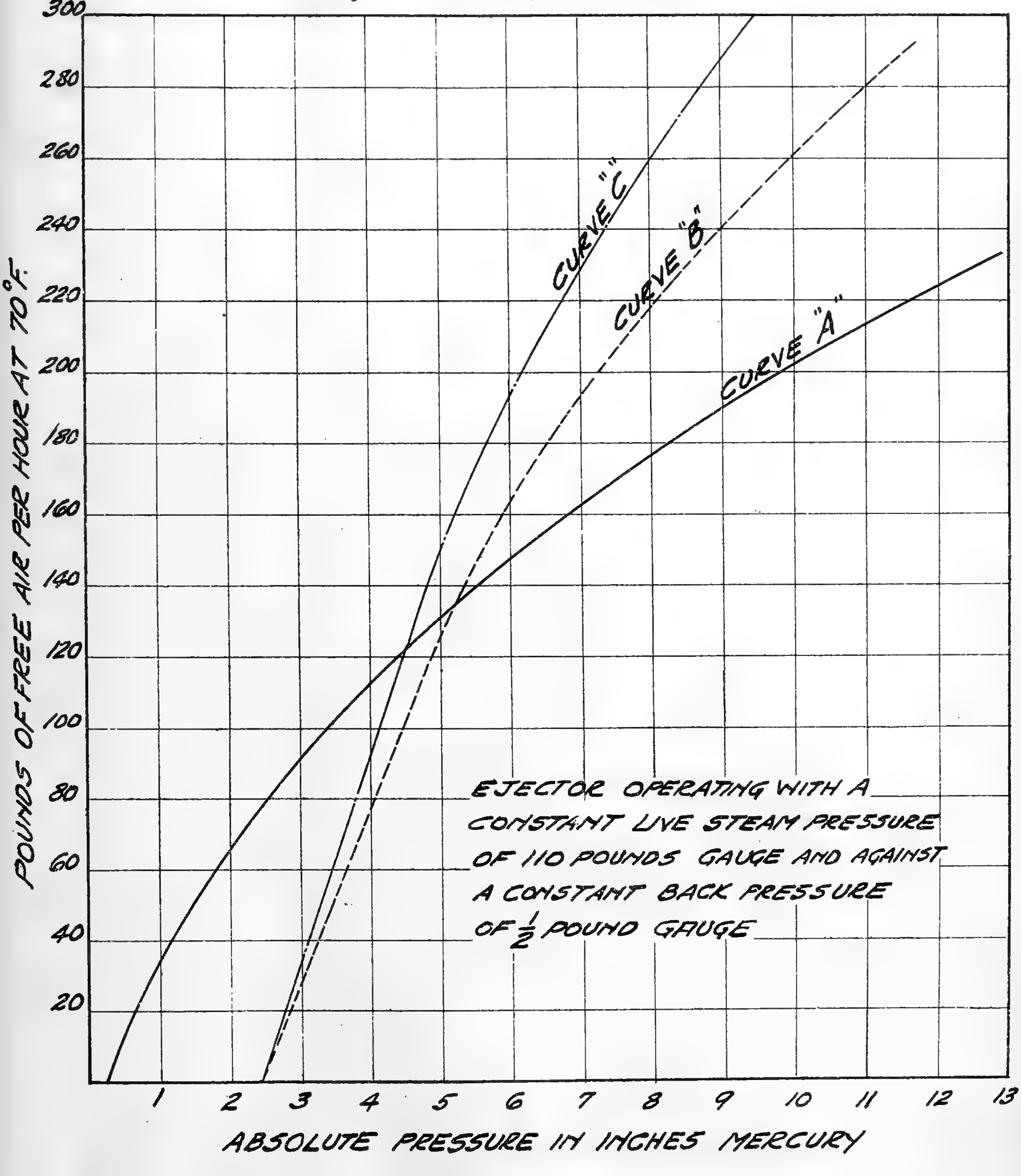
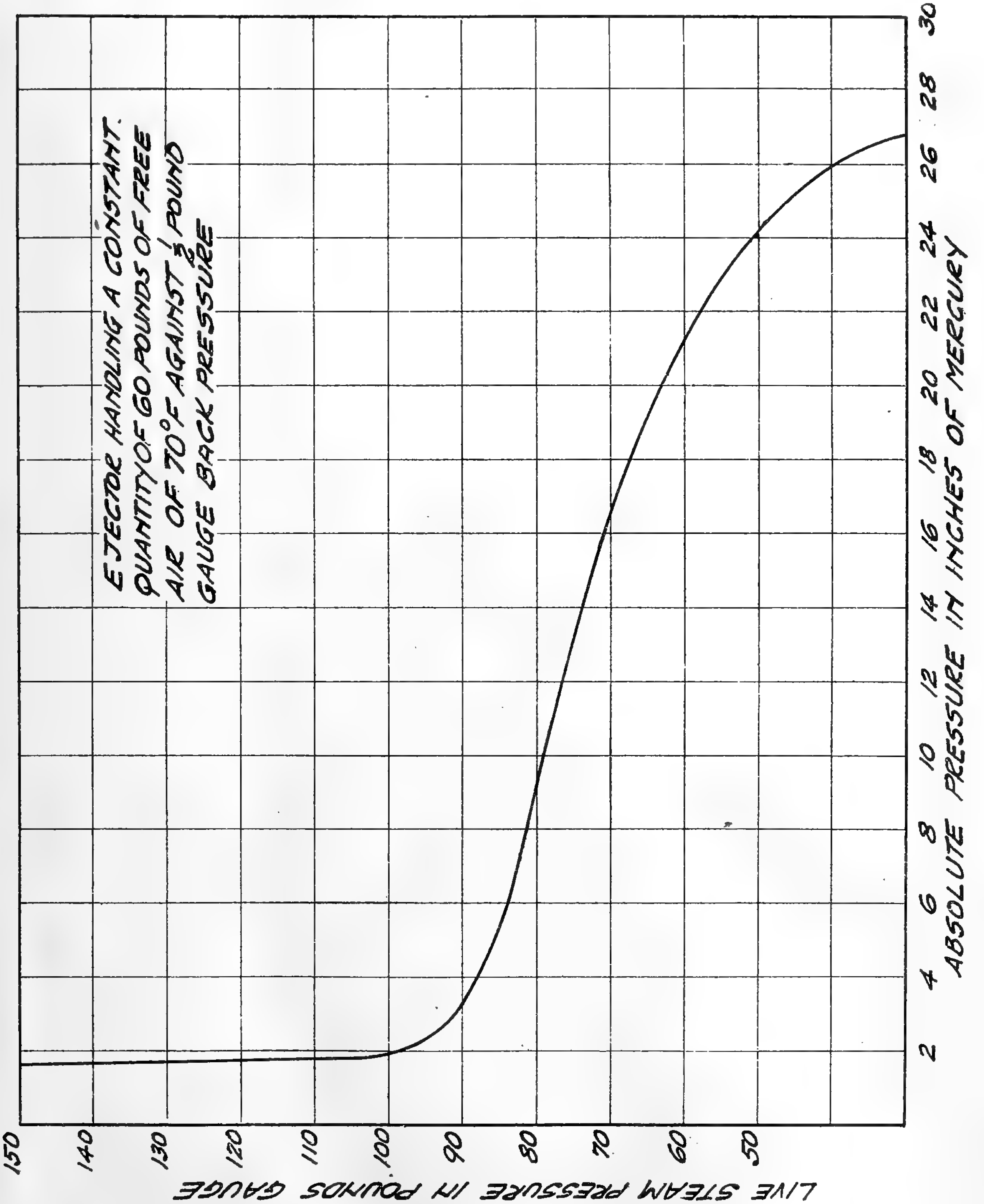
To illustrate paper on "New Developments in High Vacuum Apparatus," by G. L. E. Kothny, Esq., Member.



Curves illustrating air handling capacities of a two stage ejector at different vacua.

To illustrate paper on "New Developments in High Vacuum Apparatus," by G. L. E. Kothny, Esq., Member.



Curve illustrating the influence of variations in live steam pressure.

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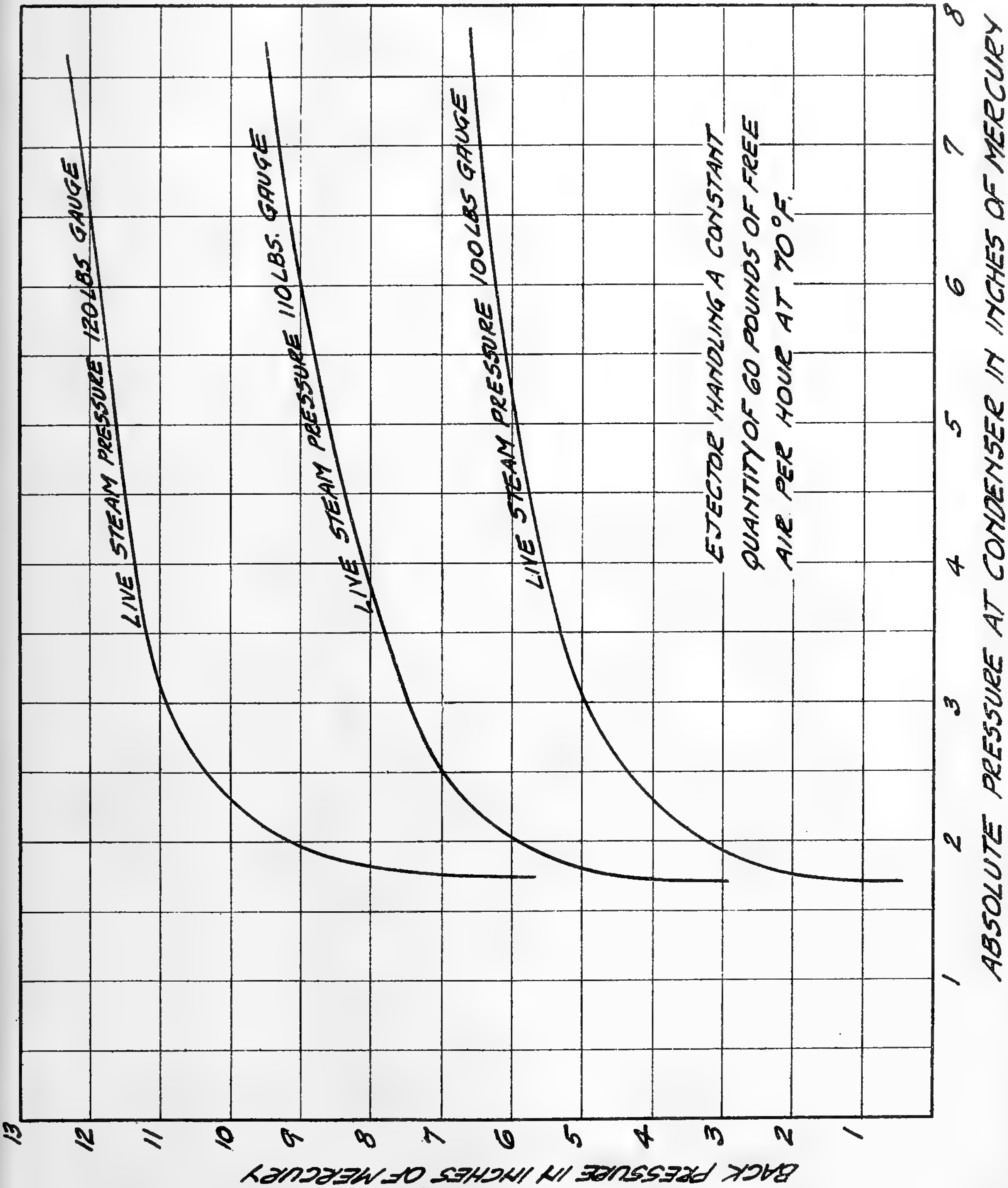
To illustrate paper on "New Developments in High Vacuum Apparatus," by G. L. E. Kothny, Esq., Member.



Curve illustrating the influence of variations in the back pressure at discharge.

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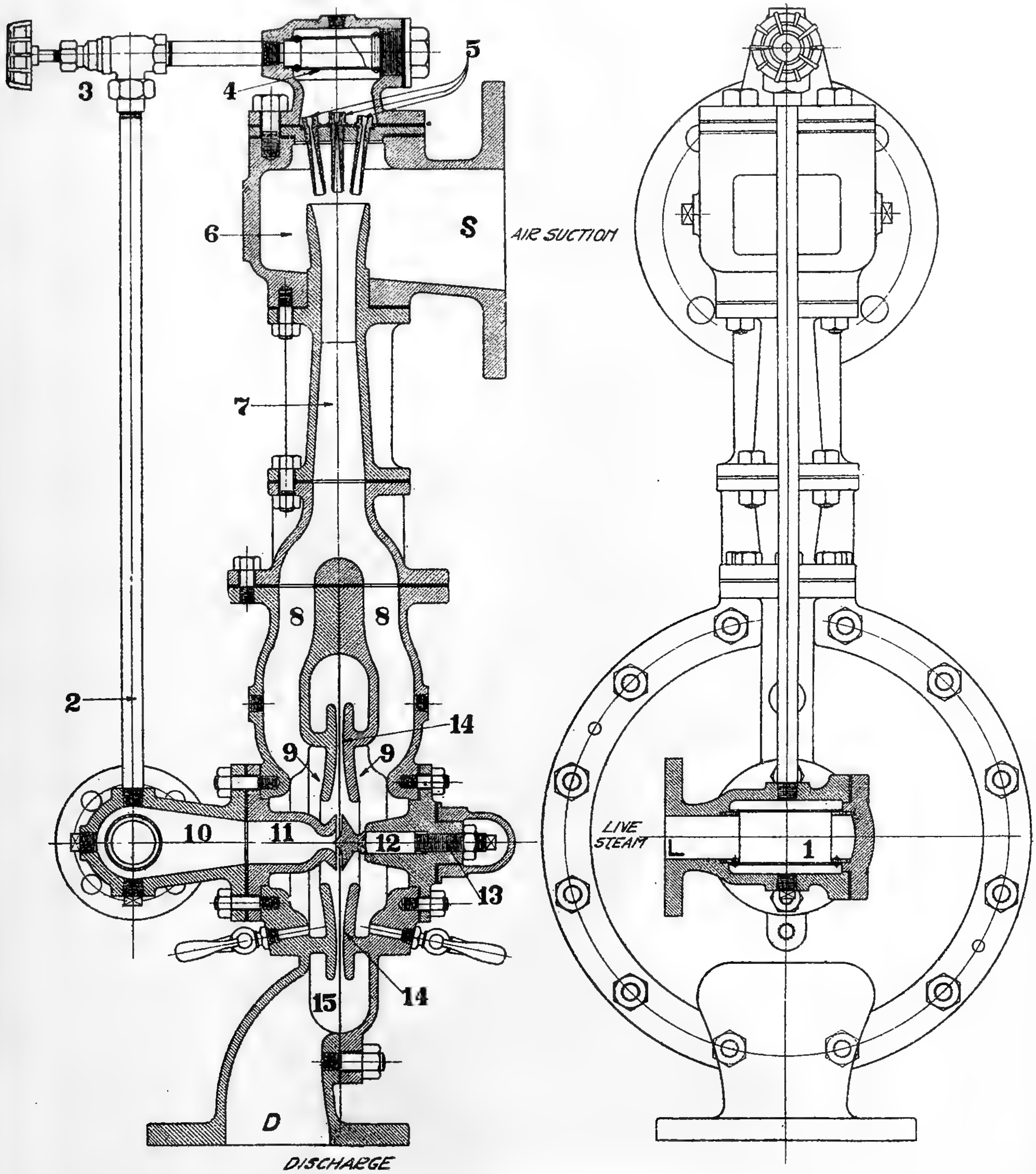

Cross section thru Radojet Air Ejector.

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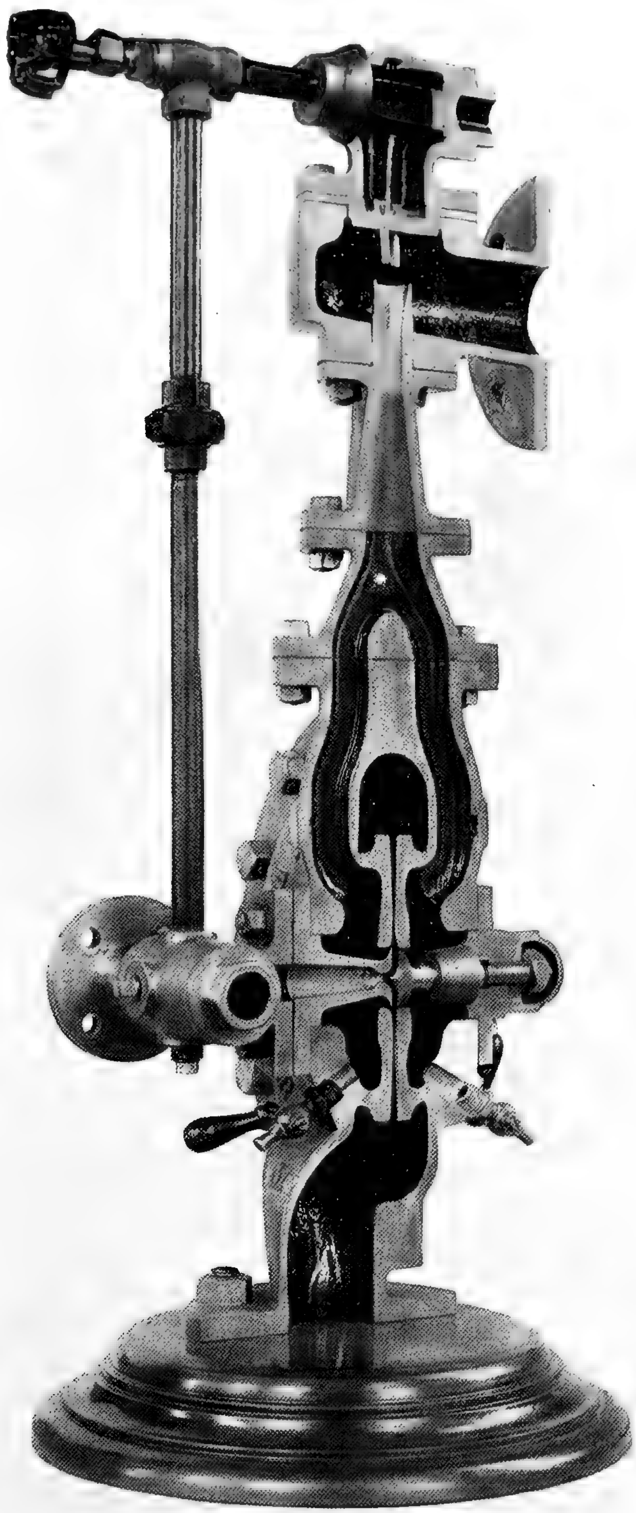
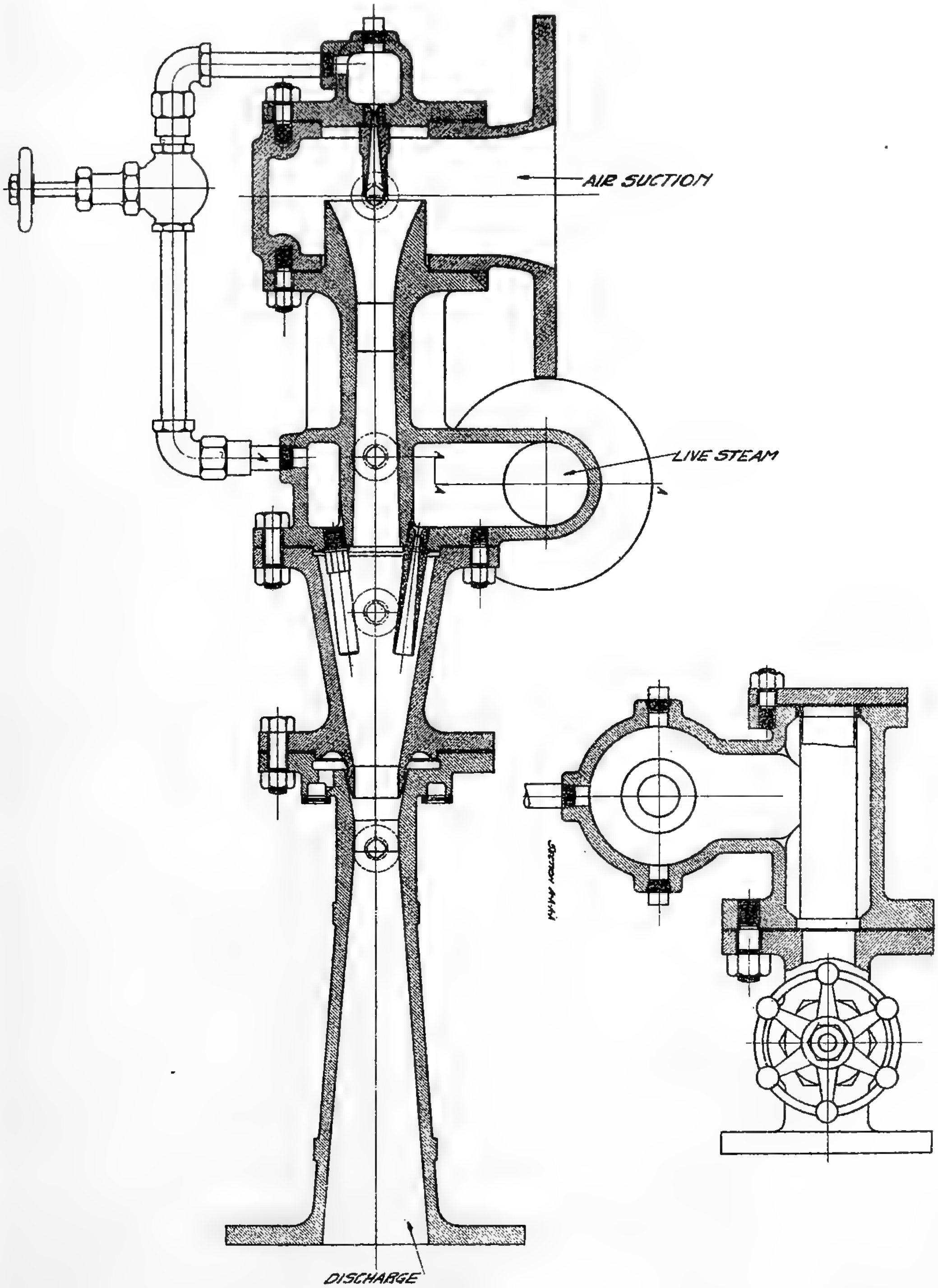
To illustrate paper on "New Developments in High Vacuum Apparatus," by G. L. Kothny, Esq., Member.



Radojet Cut in Half at Center Line.

To illustrate paper on "New Developments in High Vacuum Apparatus," by G. L. E. Kothny, Esq., Member.



Cross section through LeBlanc Air Ejector.

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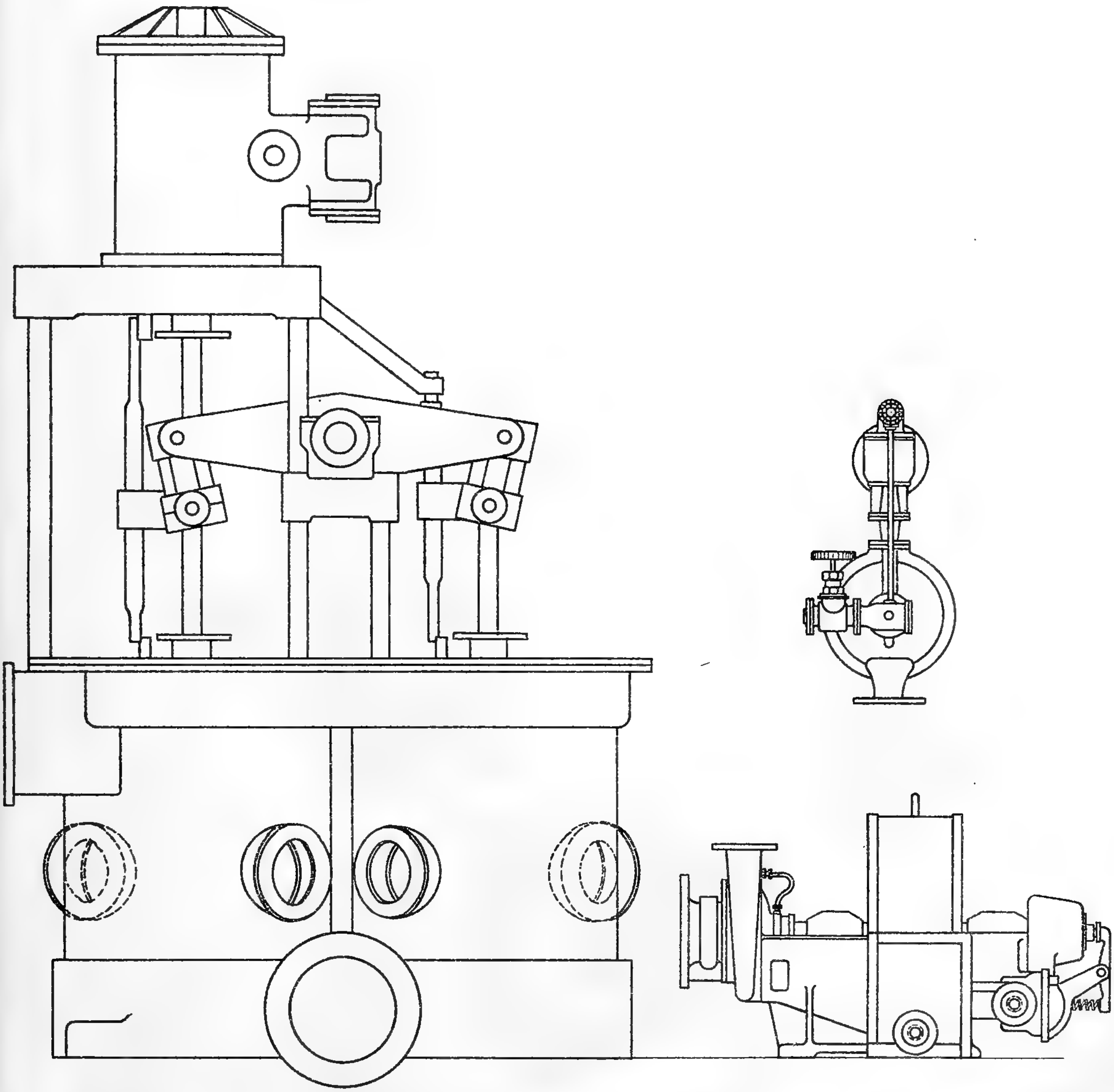
To illustrate paper on "New Developments in High Vacuum Apparatus," by G. L. E. Kothny, Esq., Member.



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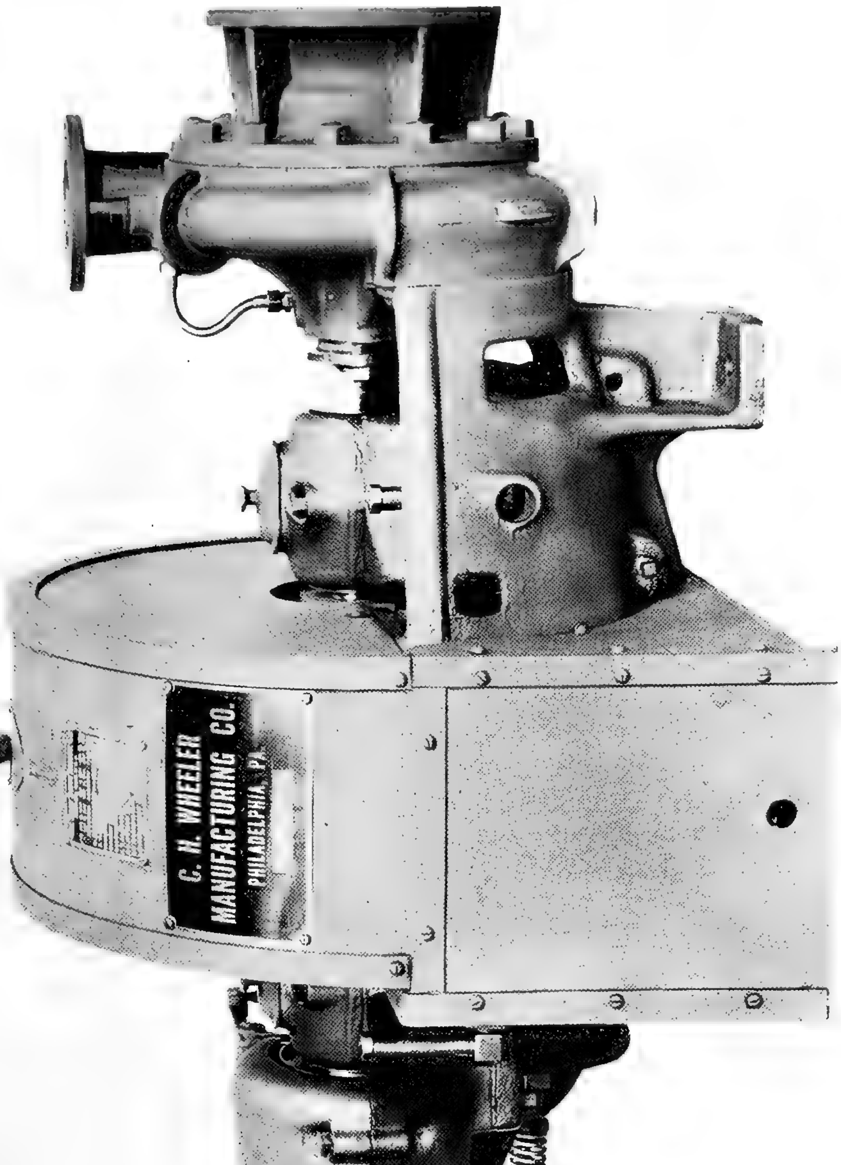
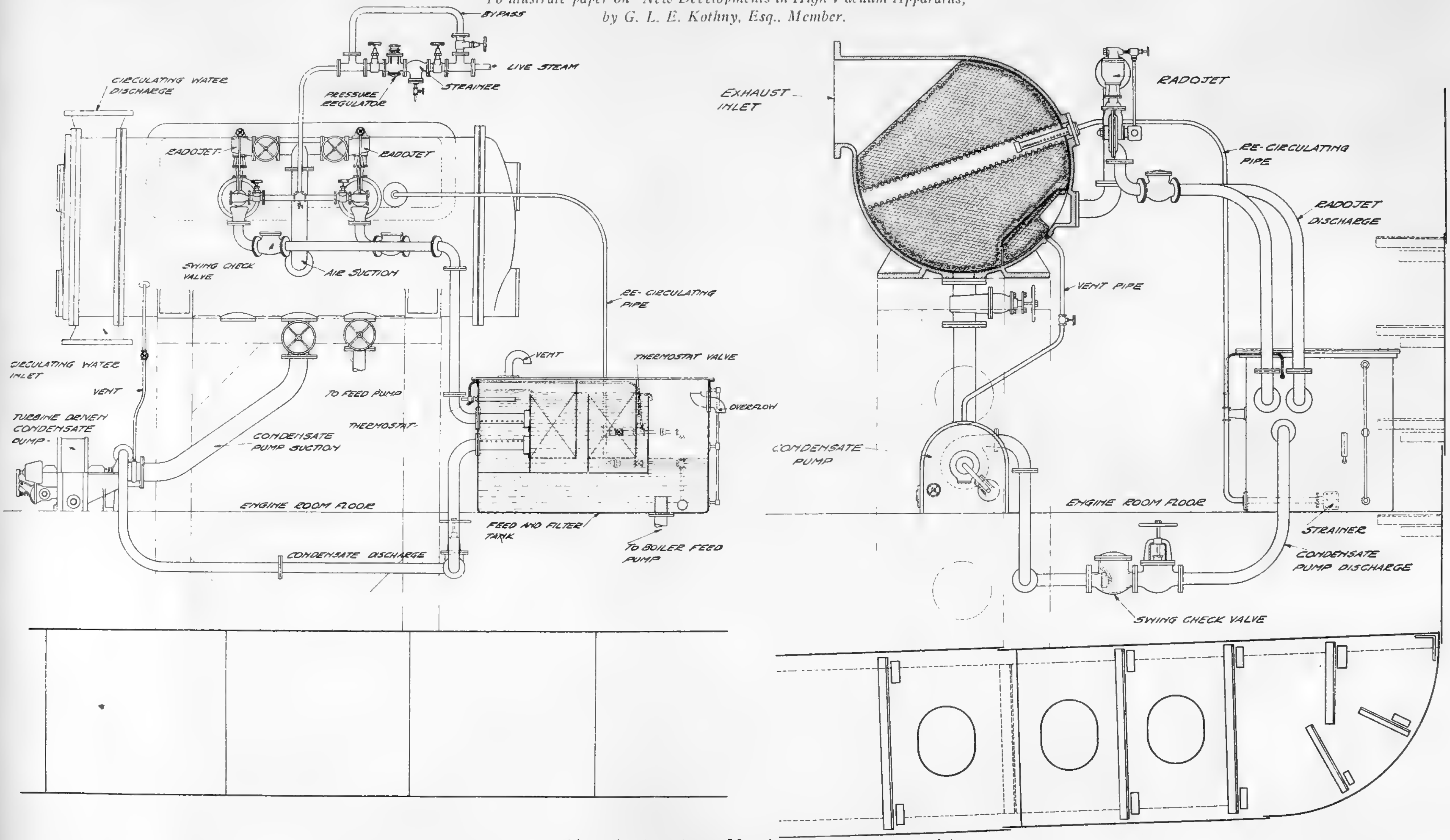
Comparative outlines of twin beam air pump and air ejector.

To illustrate paper on "New Developments in High Vacuum Apparatus," by G. L. Kothny, Esq., Member.



TURBINE-DRIVEN CONDENSATE PUMP.

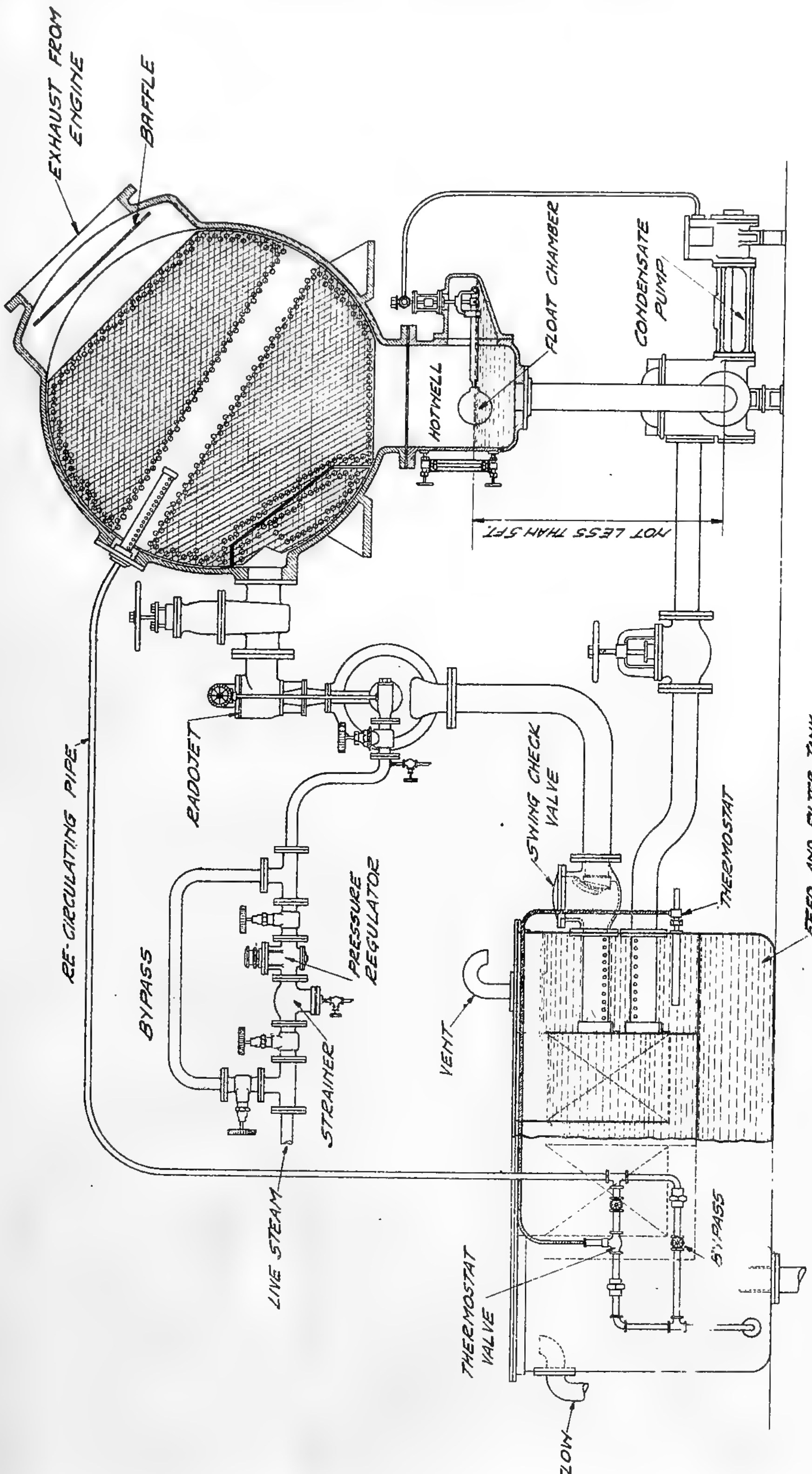
To illustrate paper on "New Developments in High Vacuum Apparatus," by G. L. E. Kothny, Esq., Member.



Air ejector installation for marine turbine surface condenser.

NORRIS PETERS INC. LITHO., WASHINGTON D.C.

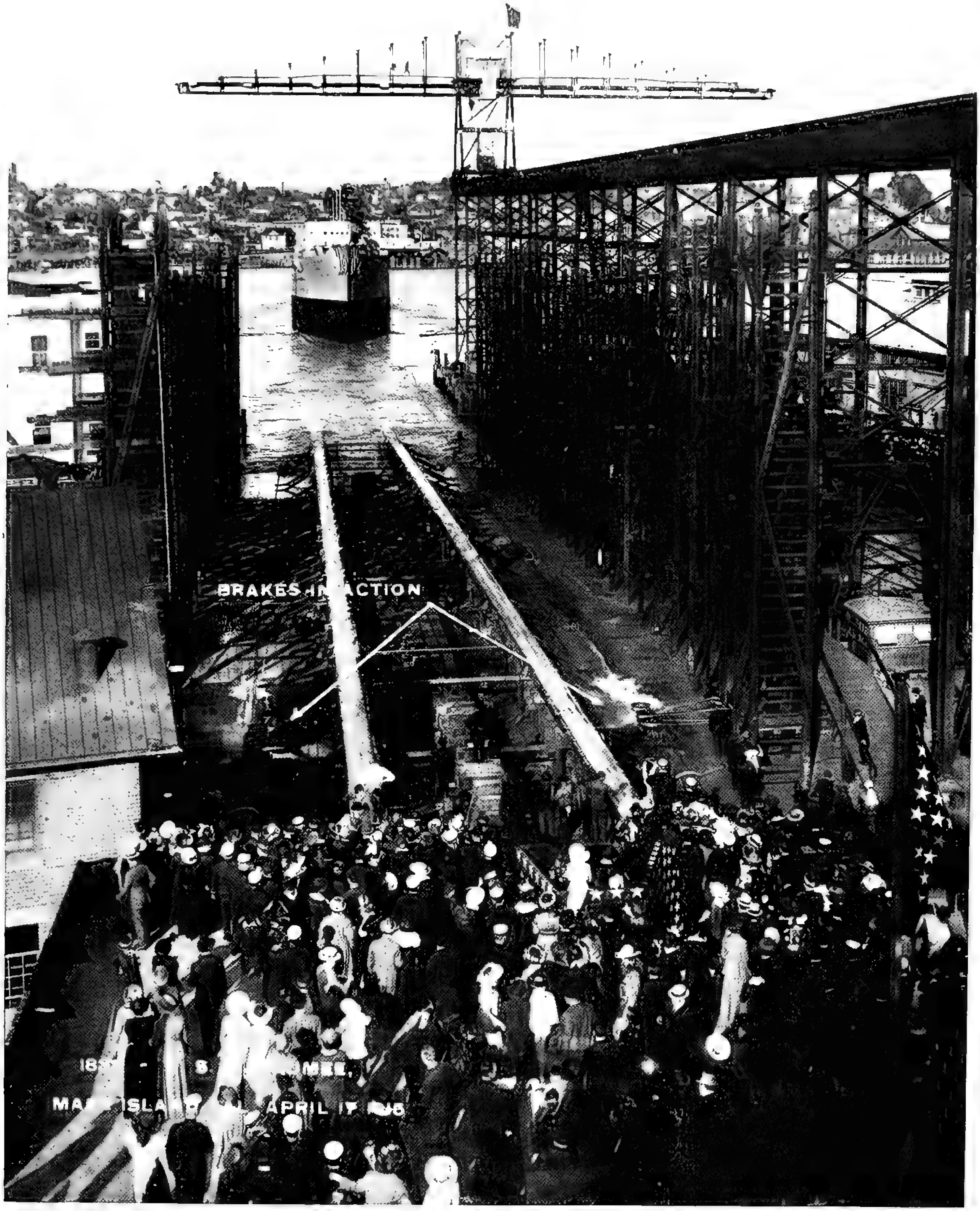
To illustrate paper on "New Developments in High Vacuum Apparatus," by G. L. E. Kothny, Esq., Member.



Air ejector installation for marine engine surface condenser.

NORRIS PETERS, INC., LITHO., WASHINGTON, D. C.

To illustrate paper on "Launching of Ships in Restricted Waters," by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.



Showing Brakes in Action in Connection With Launching of U. S. S. Maumee, Mare Island, Calif., April 17, 1915.

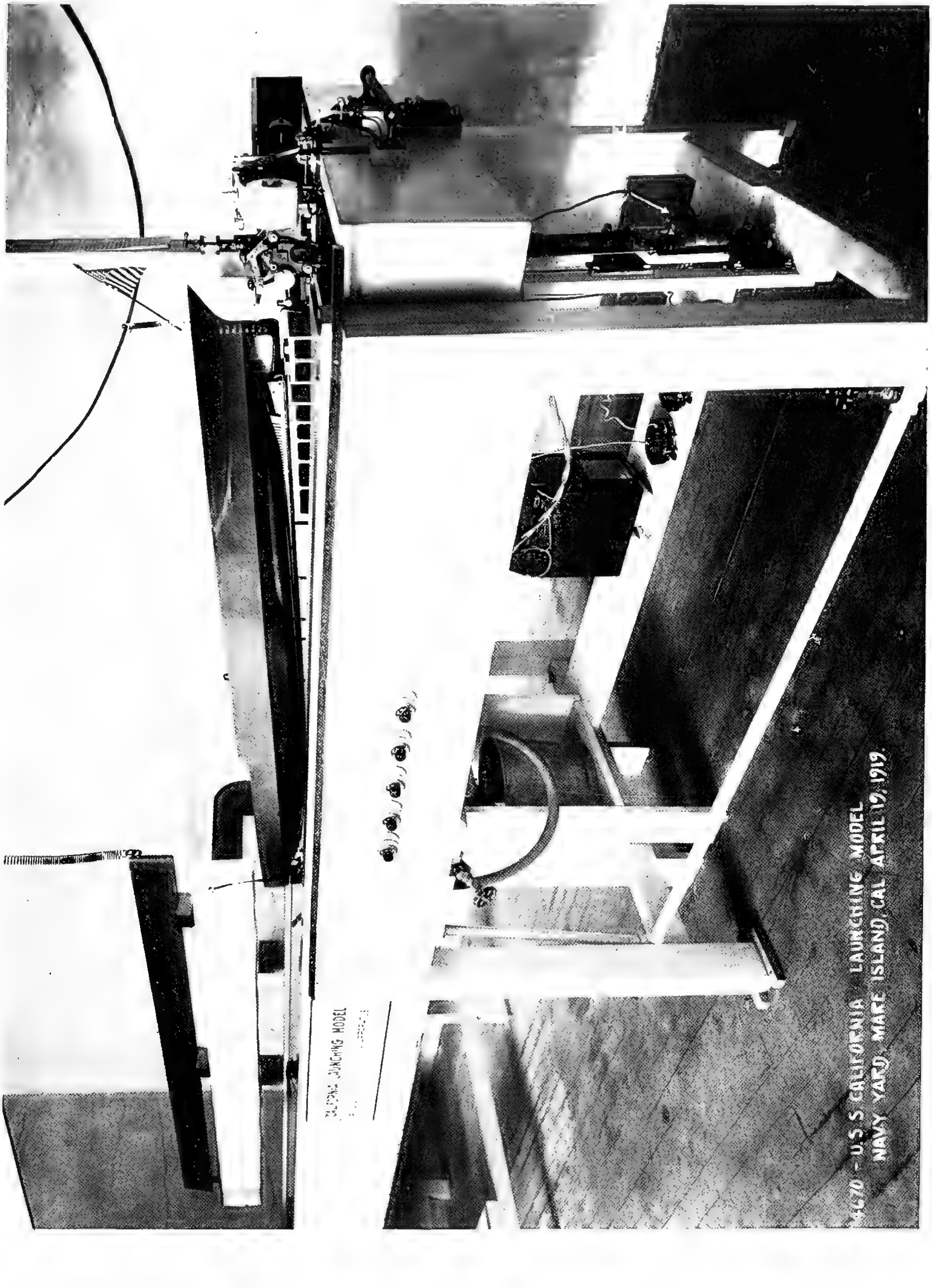
To illustrate paper on "Launching of Ships in Restricted Waters," by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.

General View of Launching Model, U. S. S. California, Mare Island, Calif., April 19, 1919.

To illustrate paper on "Launching of Ships in Restricted Waters," by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.

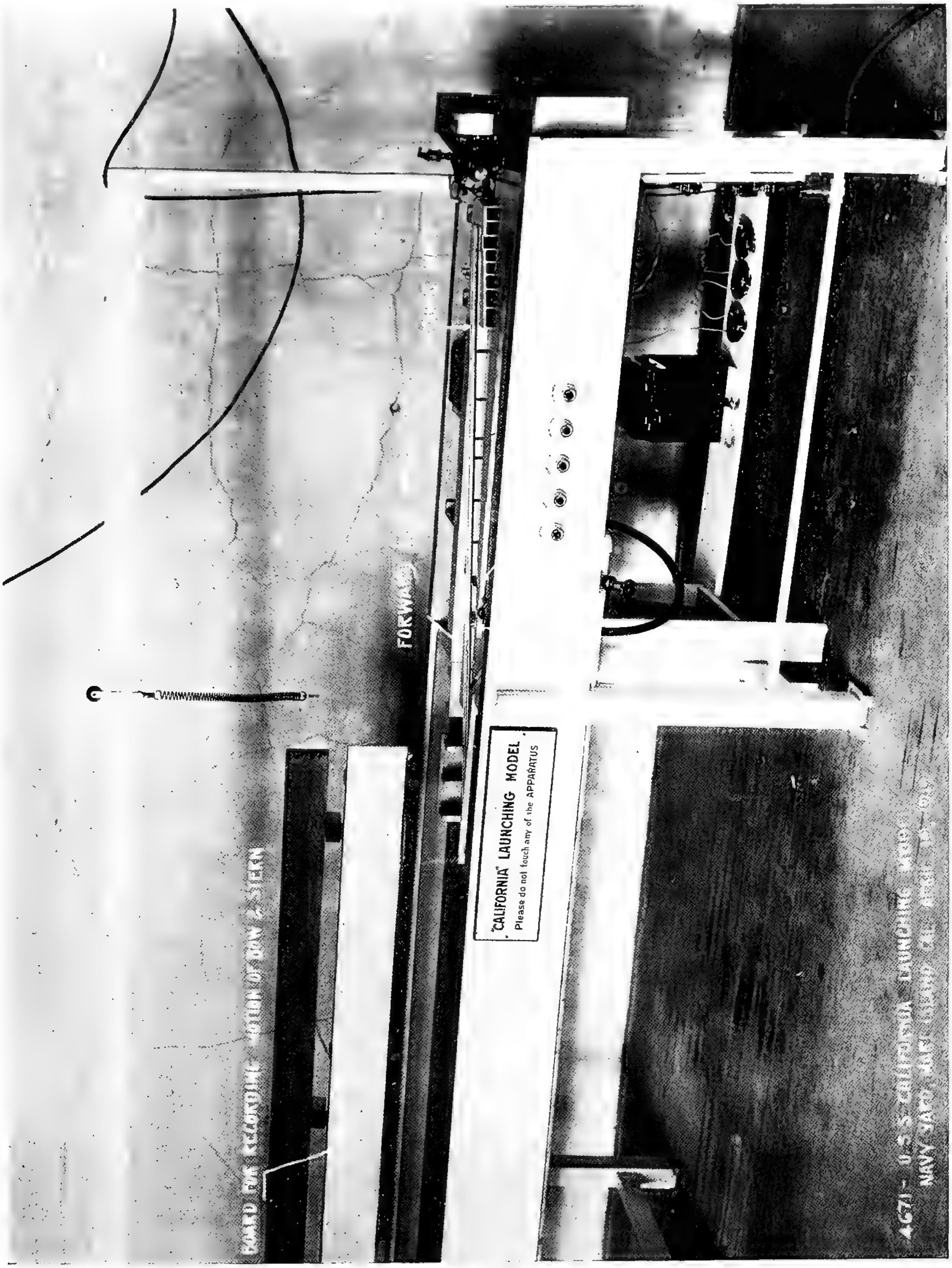
Close Stern View of Launching Model, U. S. S. California, Mare Island, Calif., April 19, 1919.

To illustrate paper on "Launching of Ships in Restricted Waters," by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.



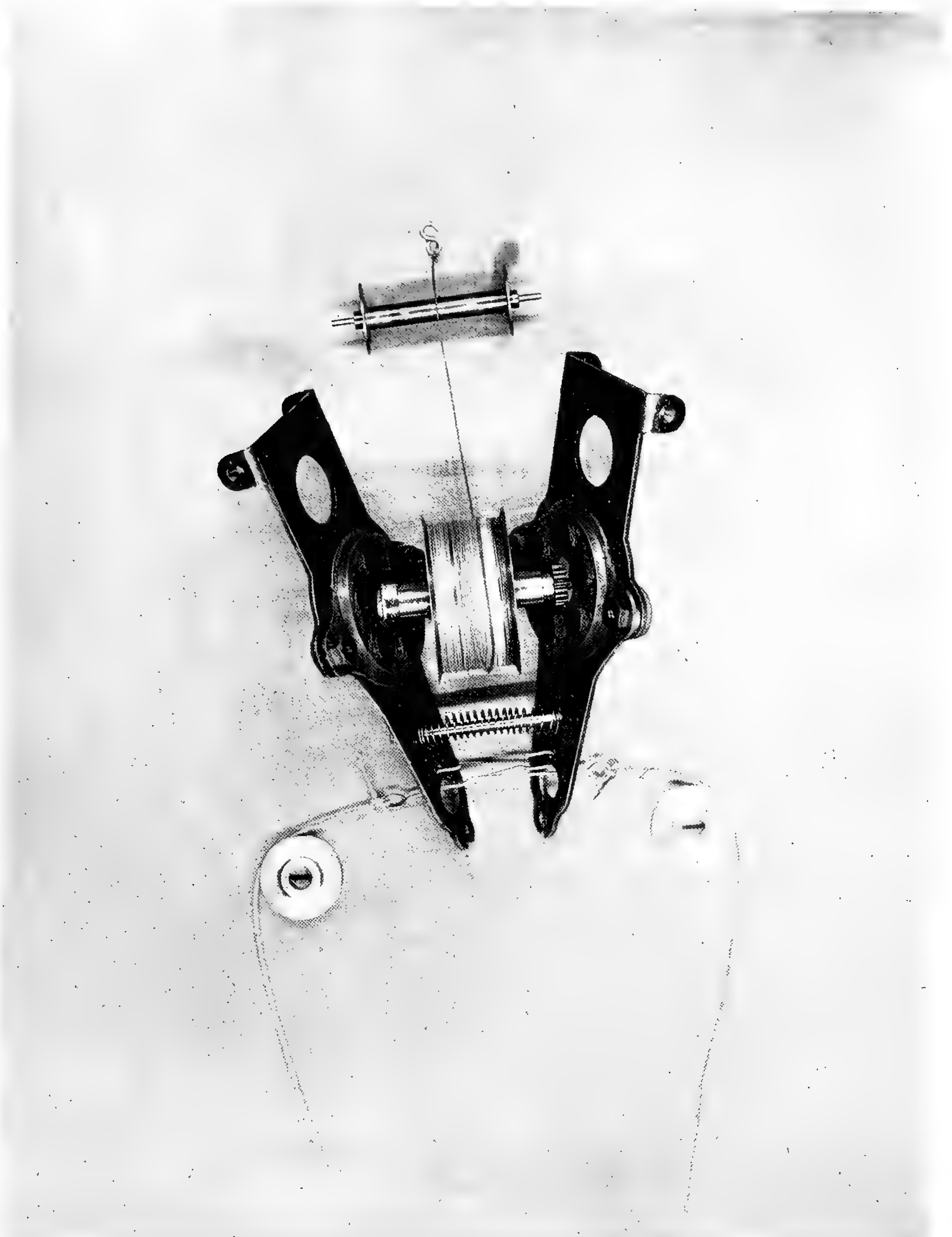
Bow View of Launching Model and Recording Mechanism, U. S. S. California, Mare Island, Calif., April 19, 1919.

To illustrate paper on "Launching of Ships in Restricted Waters," by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.



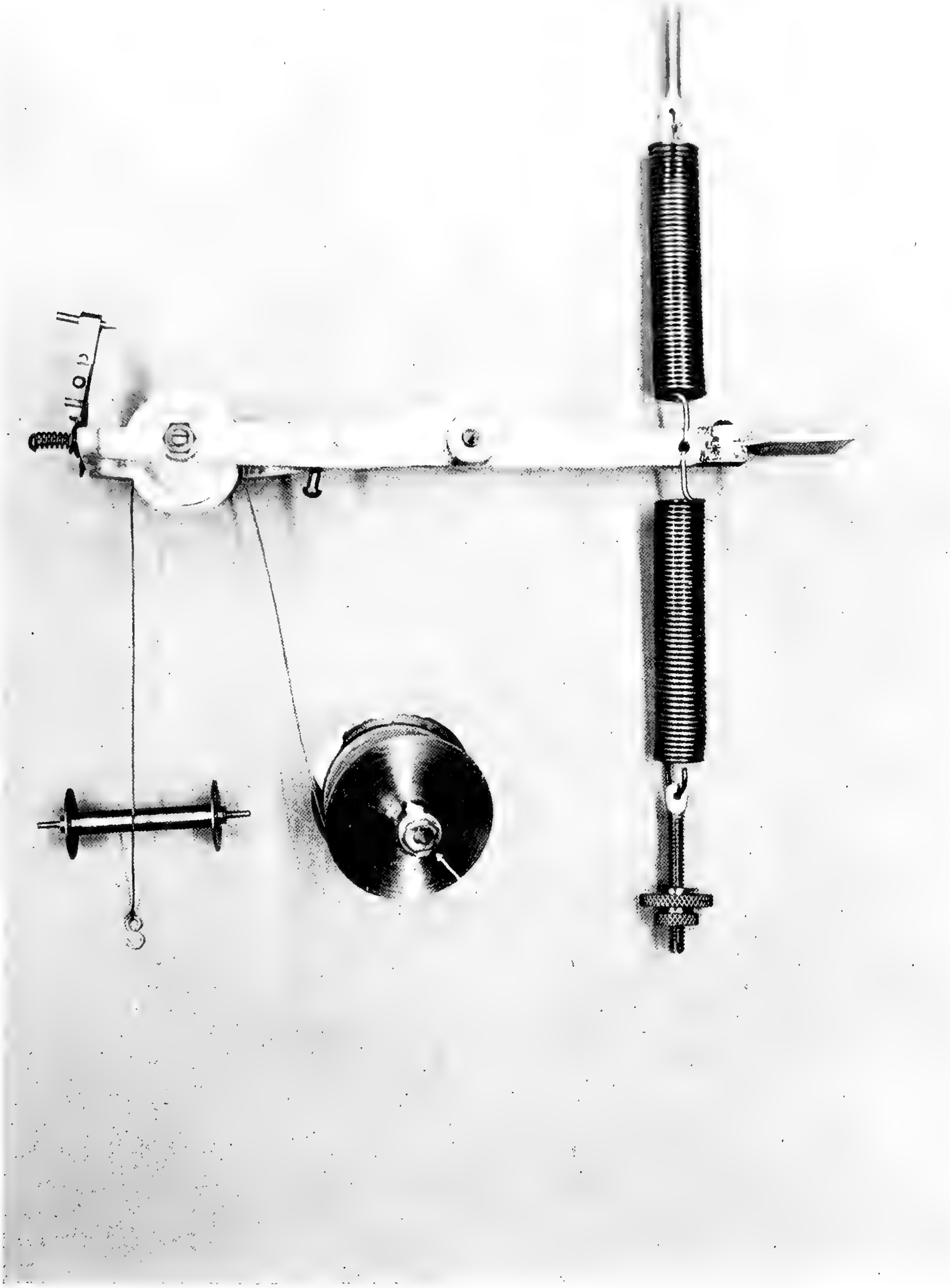
MODEL LAUNCHING WAYS AND RECORDING MECHANISM, U. S. S. CALIFORNIA, MARE ISLAND, CALIF., APRIL 19, 1919.

To illustrate paper on "Launching of Ships in Restricted Waters," by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.



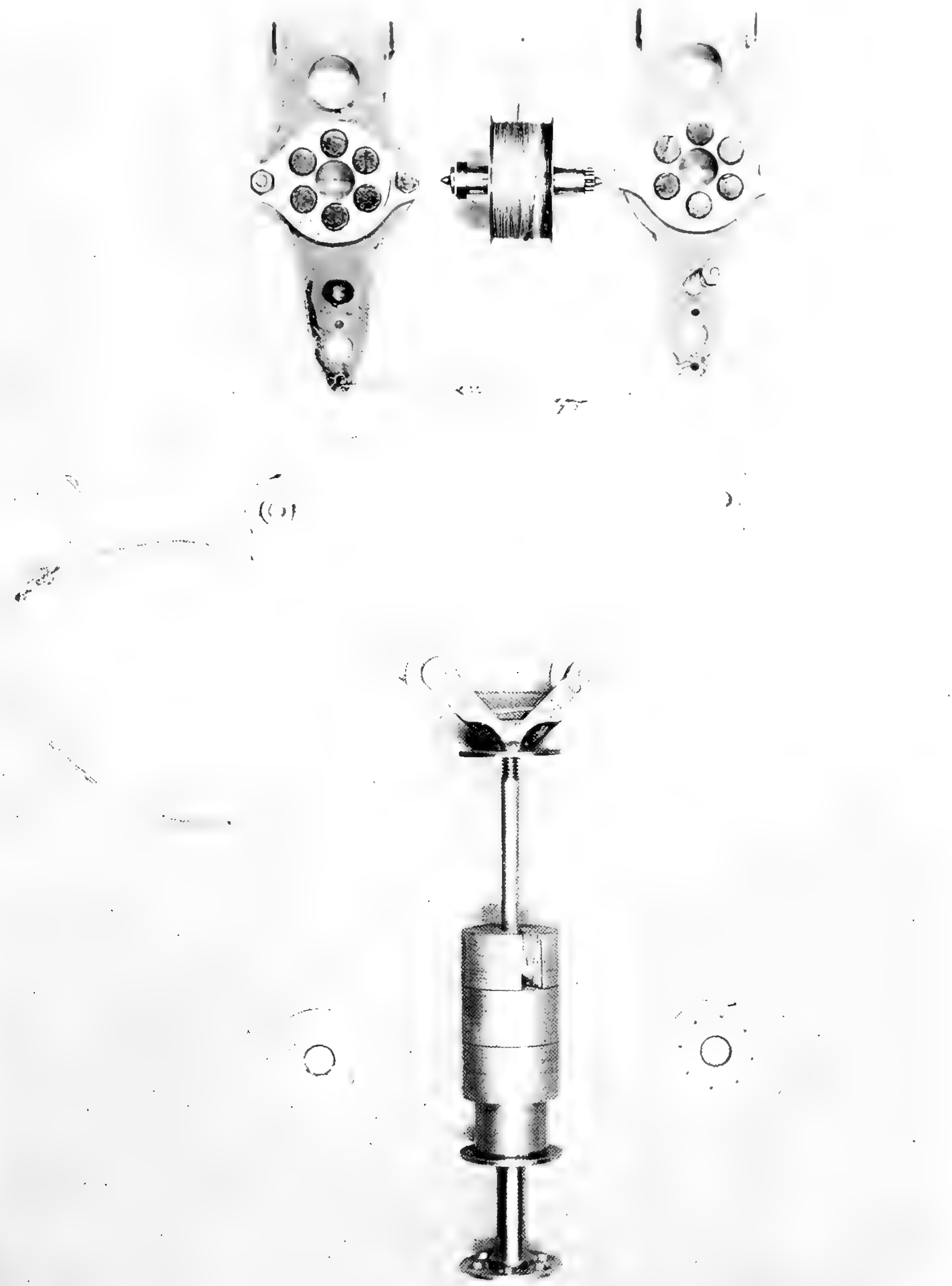
ASSEMBLY OF BRAKE GEAR, CALIFORNIA LAUNCHING MODEL.

To illustrate paper on "Launching of Ships in Restricted Waters," by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.



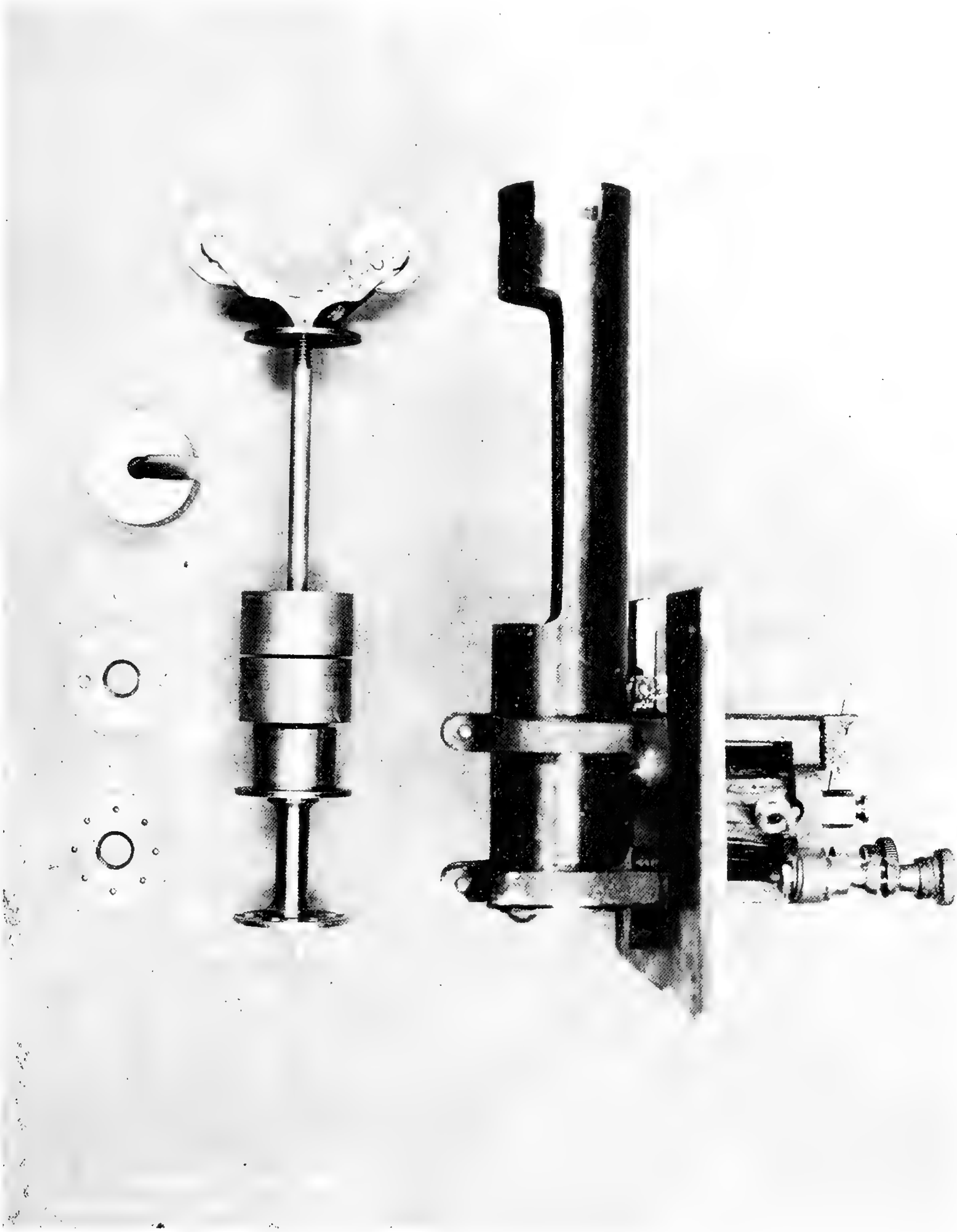
Assembly of Recording Mechanism, California Launching Model.

To illustrate paper on "Launching of Ships in Restricted Waters," by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.



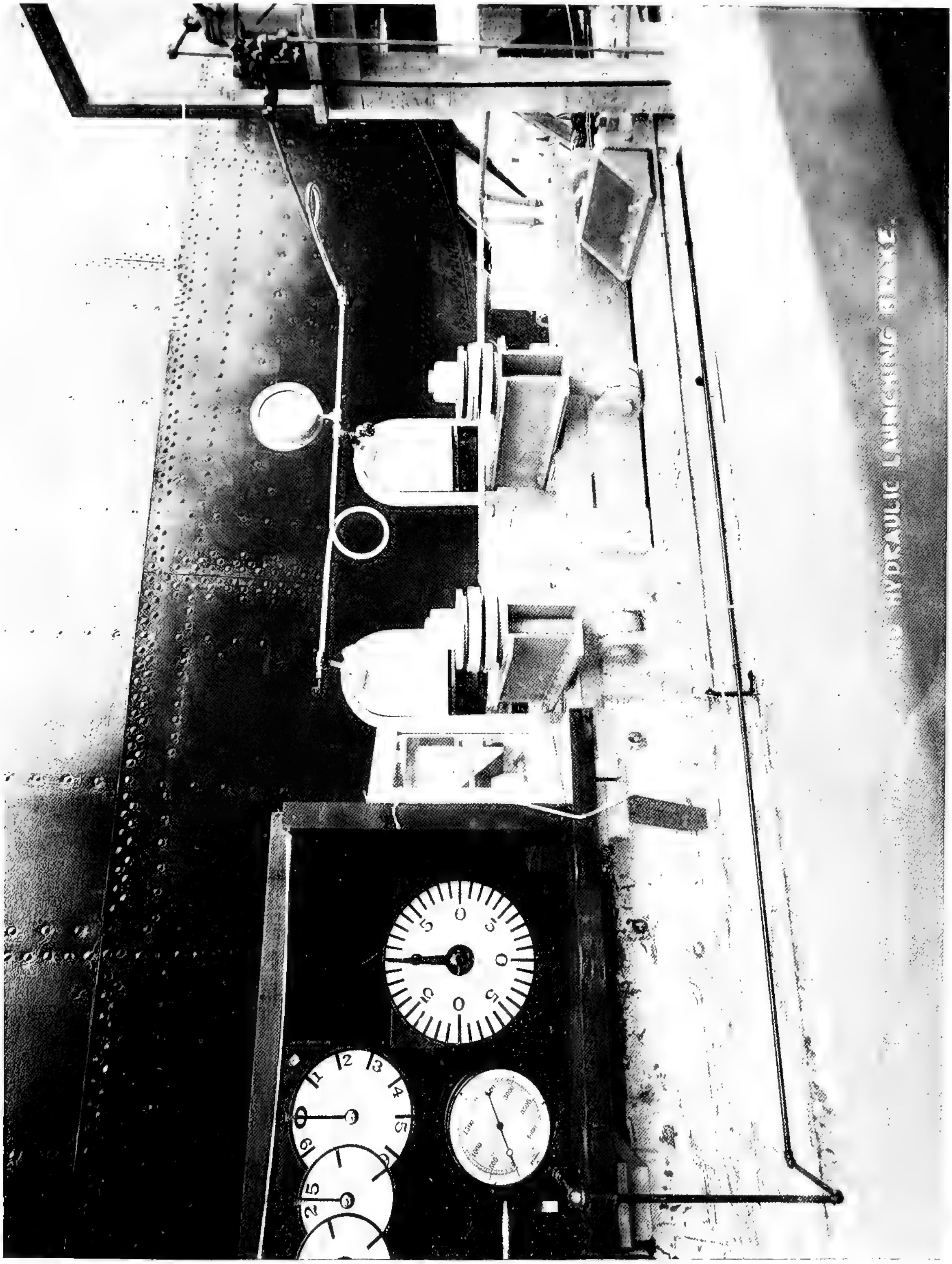
"Exploded" View of Recording Mechanism and Details of Brake, California Launching Model.

To illustrate paper on "Launching of Ships in Restricted Waters," by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.



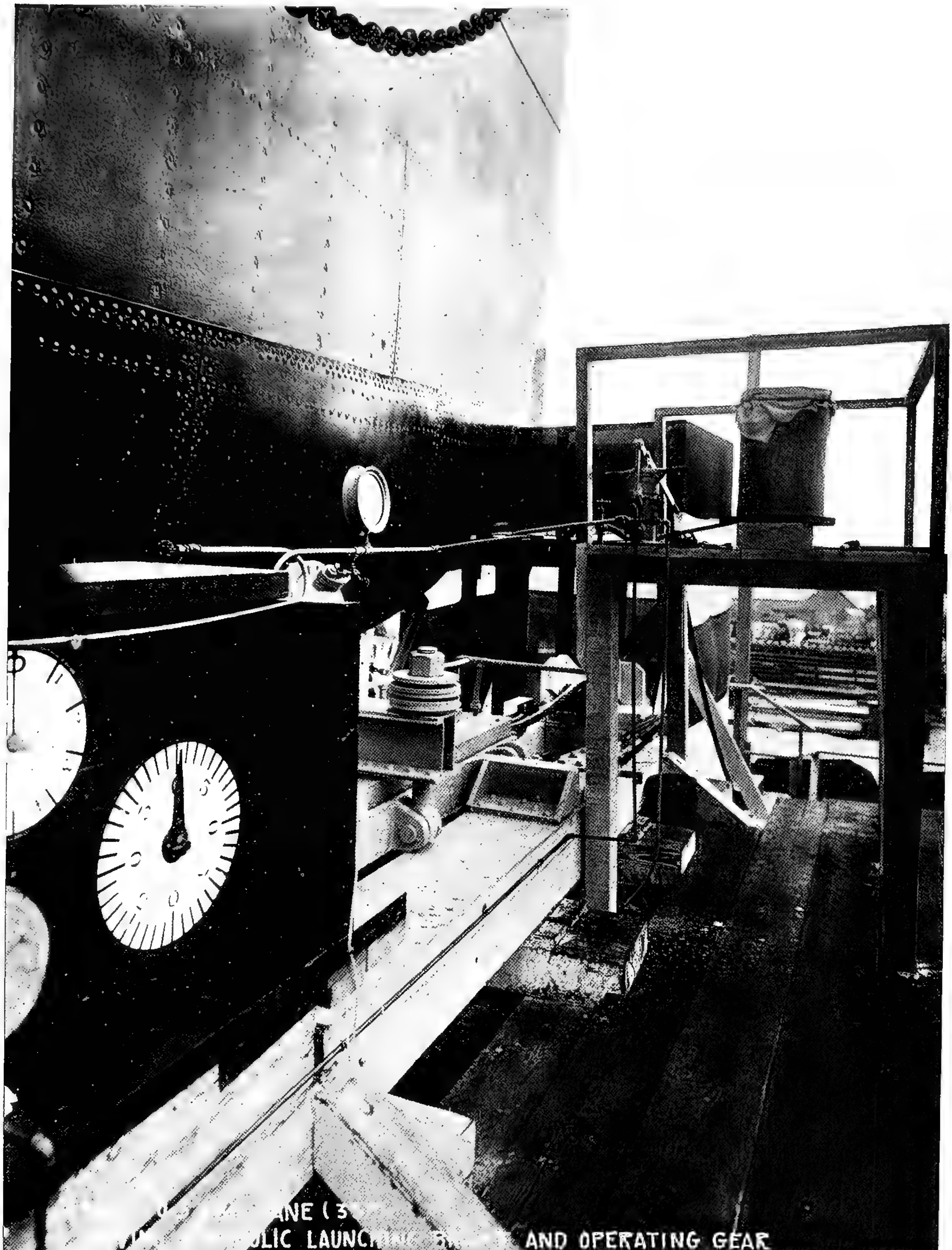
Details of Launching Brake, California Launching Model.

To illustrate paper on "Launching of Ships in Restricted Waters," by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.



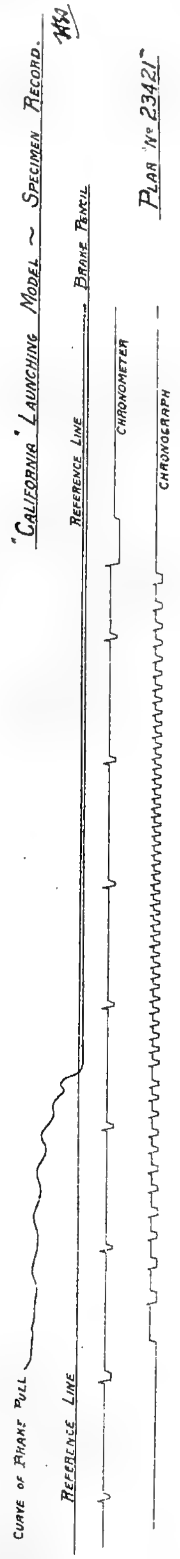
Hydraulic Launching Brake and Direct Reading Chronograph Tested Out in Launching Destroyer Zane.

To illustrate paper on "Launching of Ships in Restricted Waters," by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.



HYDRAULIC LAUNCHING BRAKE TESTED OUT IN LAUNCHING DESTROYER ZANE.

To illustrate paper on "Launching of Ships in Restricted Waters," by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander H. E. Saunders, Construction Corps, U. S. N., Member.



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To illustrate paper on "Launching of Ships in Restricted Waters."
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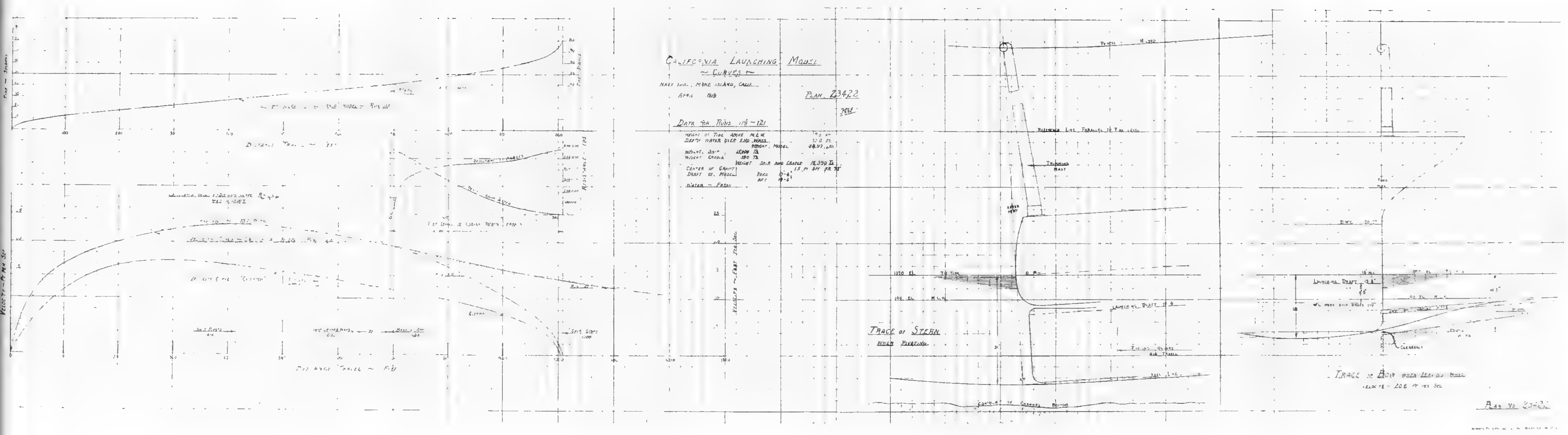
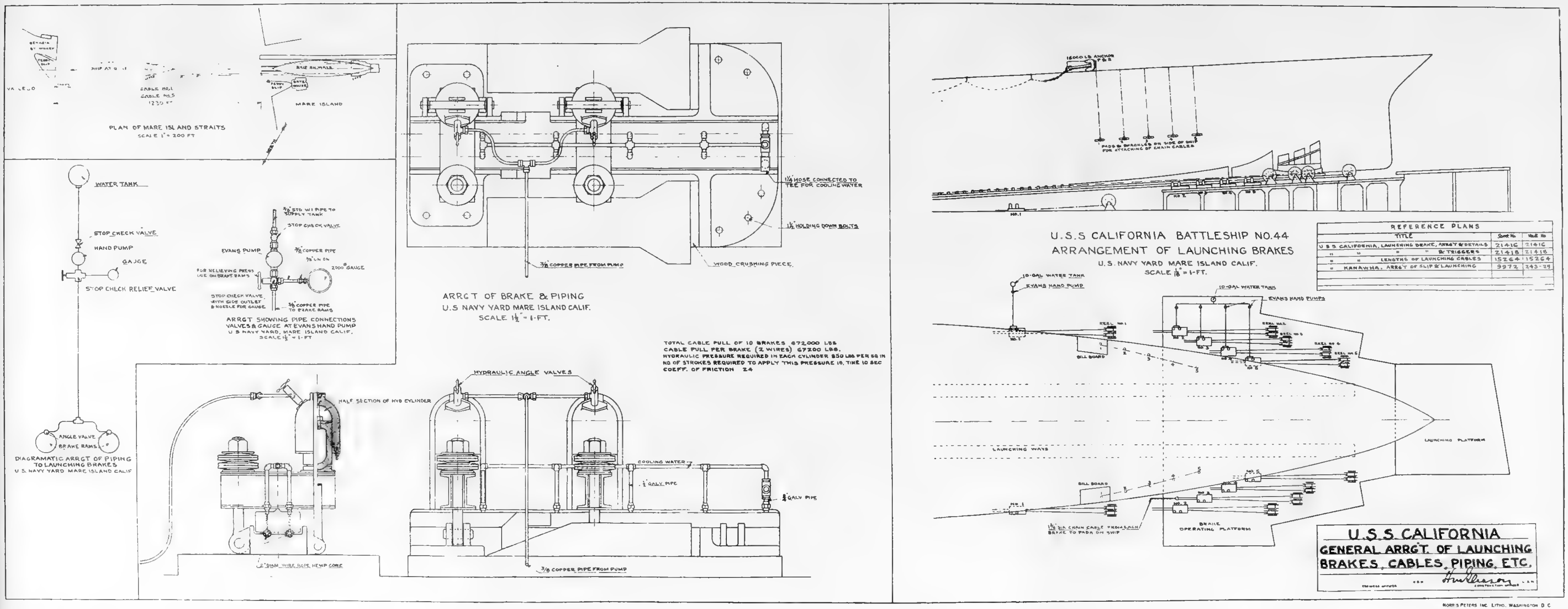
H. E. Saunders, Construction Corps, U. S. N., Member.



[illegible] paper on "Launching of
[illegible], Construction Corps, [illegible]
[illegible]

To illustrate paper on "Launching of Ships in Restricted Waters."
by Captain H. M. Gleason, Construction Corps, U. S. N., Member, and Lieutenant Commander
H. E. Saunders, Construction Corps, U. S. N., Member.



TITLE	Sheet No.	Vault No.
U.S.S. CALIFORNIA, LAUNCHING BRAKE, ARRG'T & DETAILS	21416	21416
" " " & TRIGGERS	21418	21418
" " LENGTHS OF LAUNCHING CABLES	15264	15264
" KANAWHA, ARRG'T OF SLIP & LAUNCHING	9972	243-29

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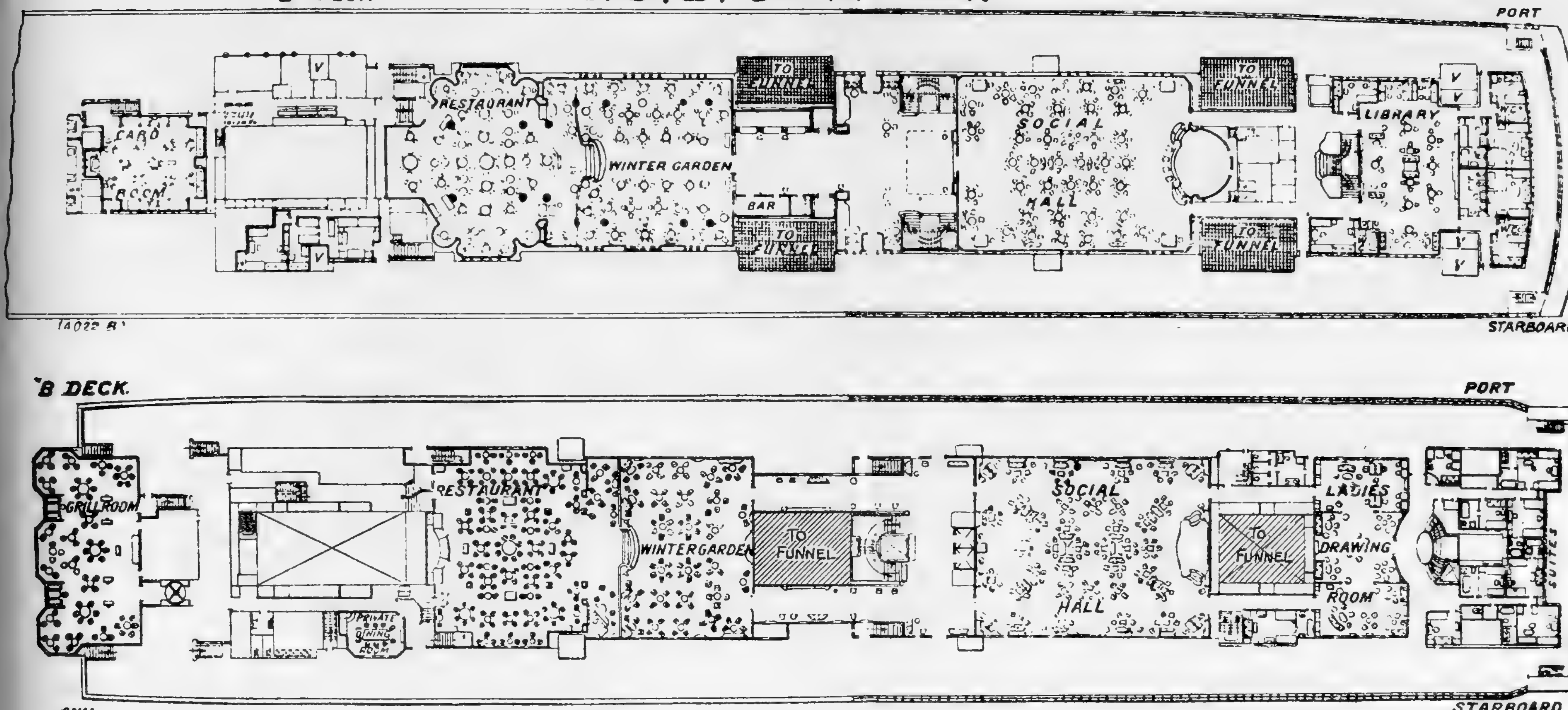
To illustrate paper on "The Propelling Machinery of the U. S. S. Leviathan,"
by Ernest H. B. Anderson, Member.

U.S.S. LEVIATHAN.



S.S. IMPERATOR. PLAN OF "B" DECK.

Fig. I.

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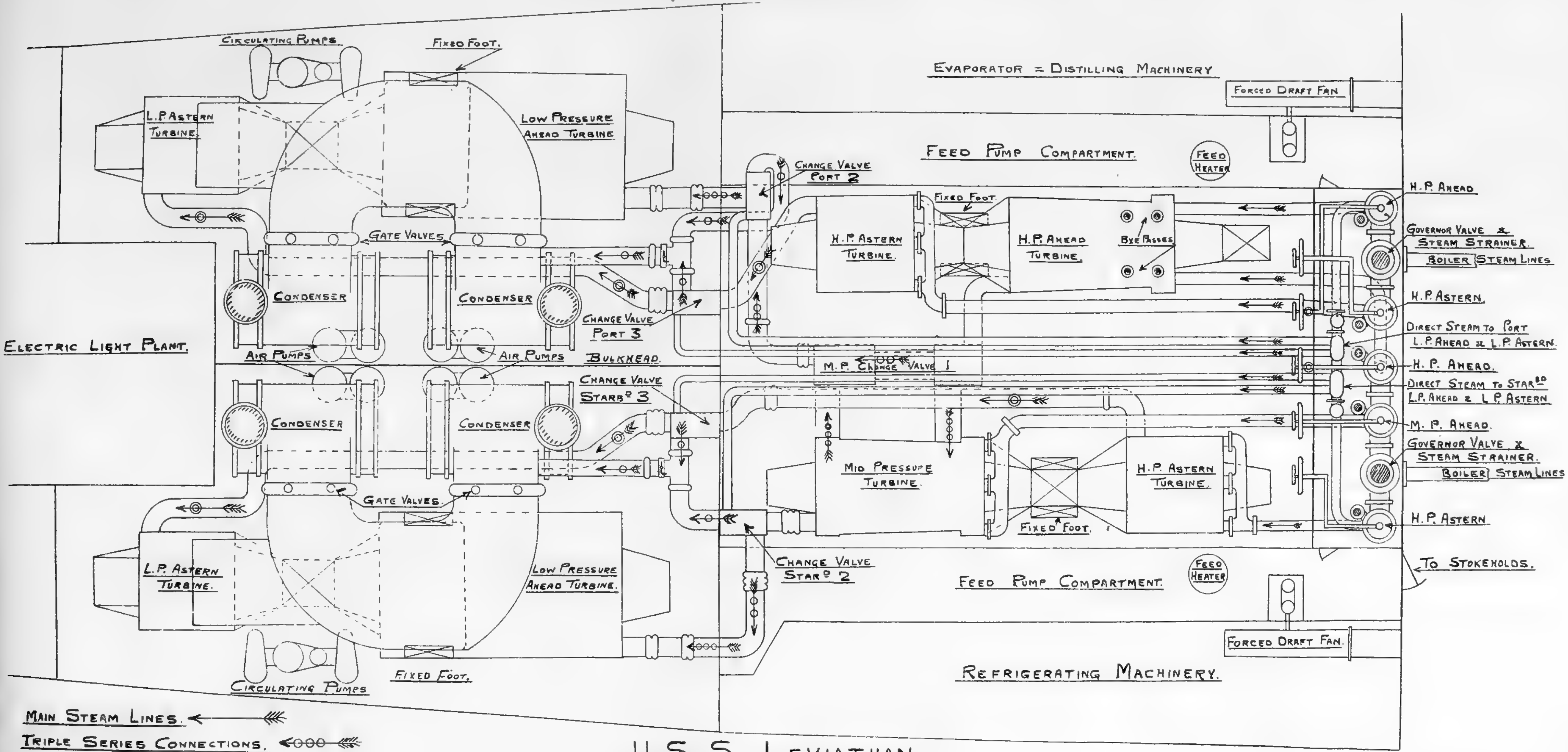

Main Steam Lines.
Triple Series Connections.
Manoeuvre Connections.
Single Connections.
Astern Connections.

U. S. S. Leviathan.

Diagram of Parsons Turbine Machinery with Steam & Exhaust Connections.

Fig. 2.

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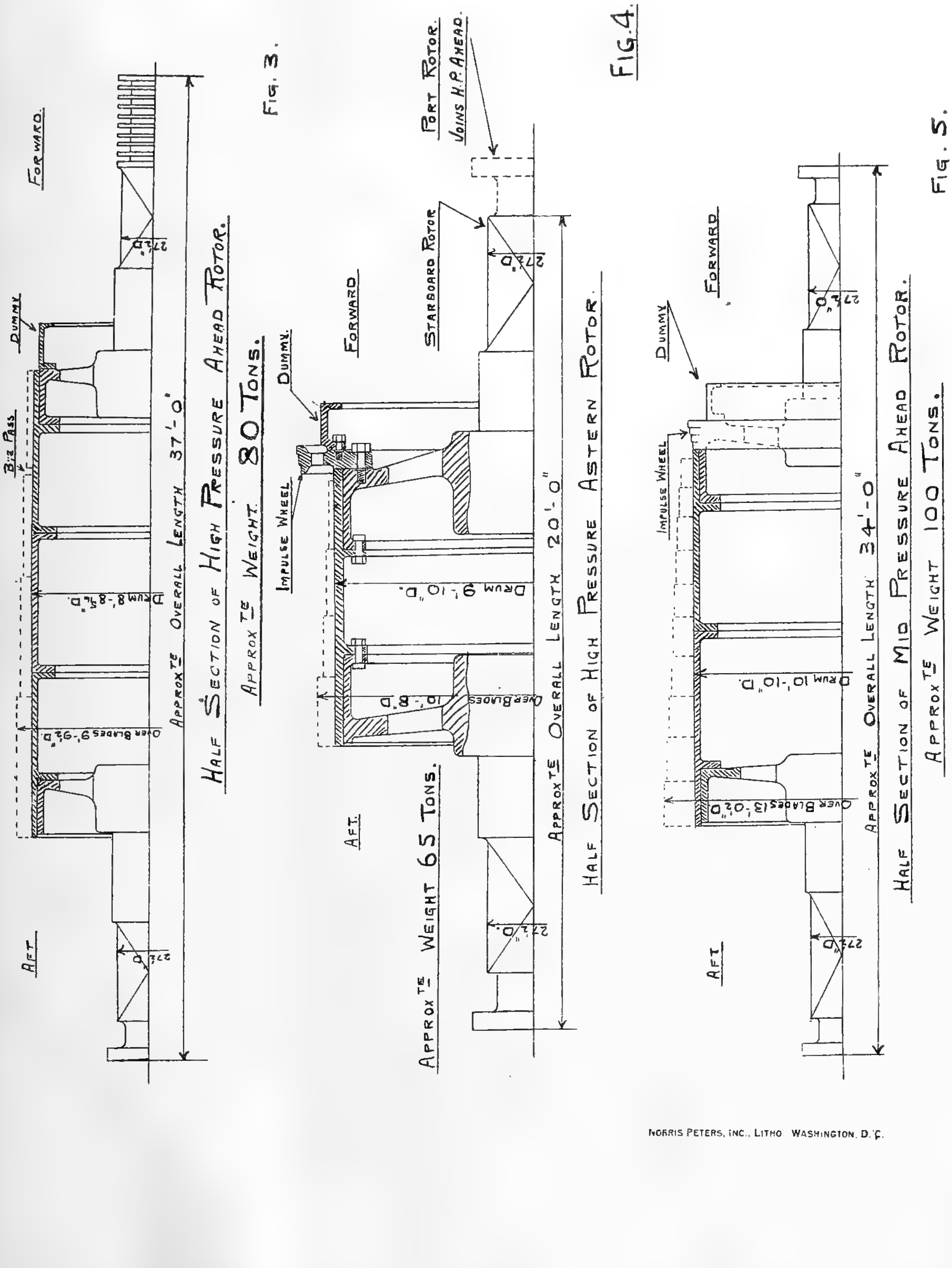
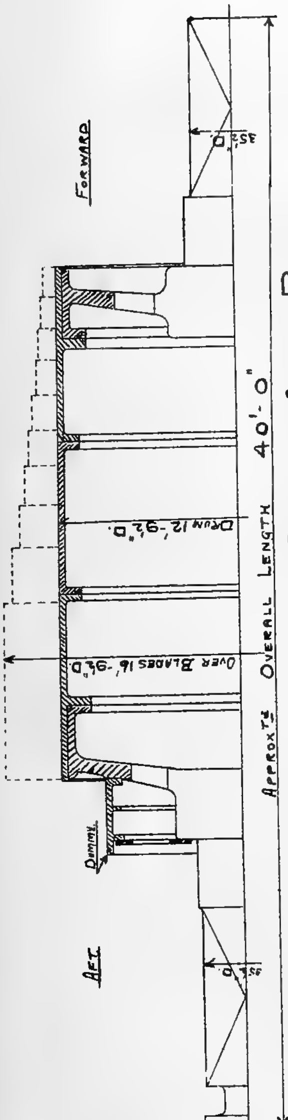
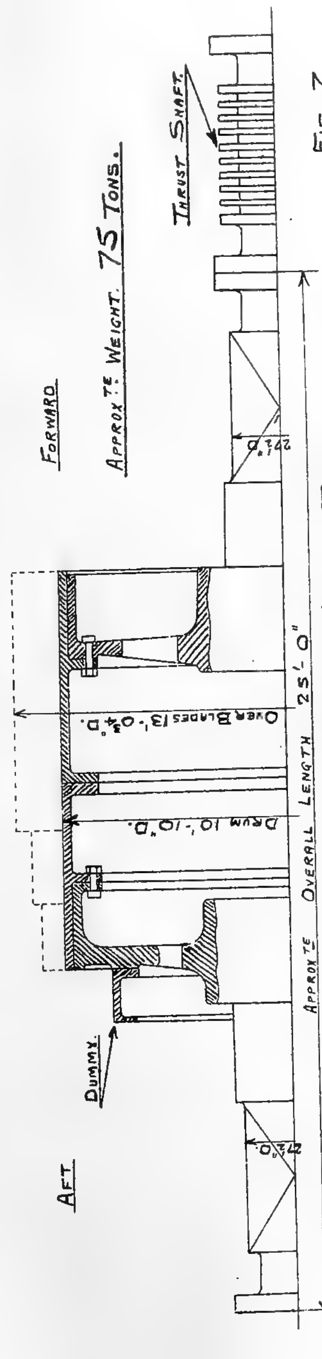


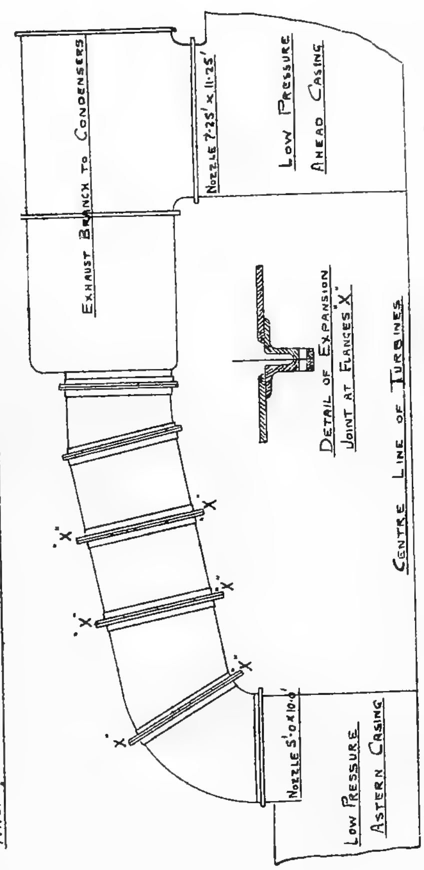
To illustrate paper on "The Propelling Machinery of the U. S. S. Leviathan," by Ernest H. B. Anderson, Member.



Half Section of Low Pressure Ahead Rotor.
Approx'te Weight 150 Tons.
Fig. 6.



Approx'te Weight 75 Tons.
Half Section of Low Pressure Astern Rotor.
Fig. 7.



Side Elevation of Exhaust Arch between Low Press. Astern & Nozzle to Condensers.
Fig. 8.

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by Ernest H. B. Anderson, Member.

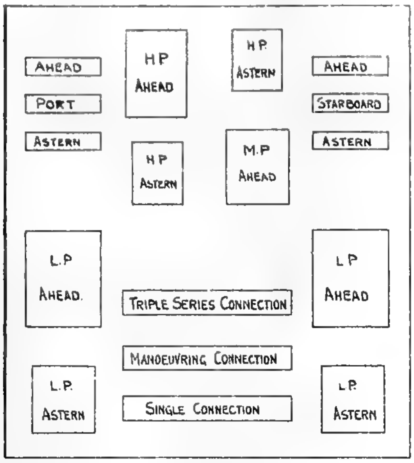
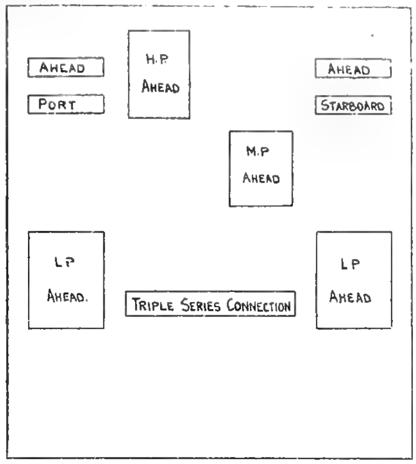
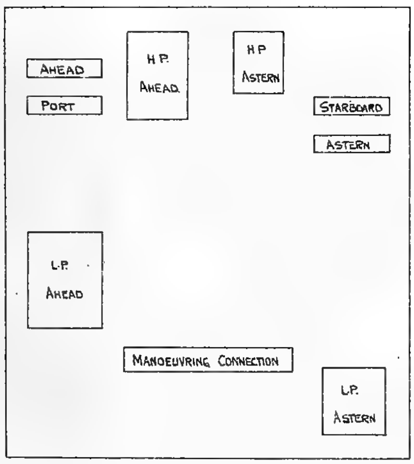


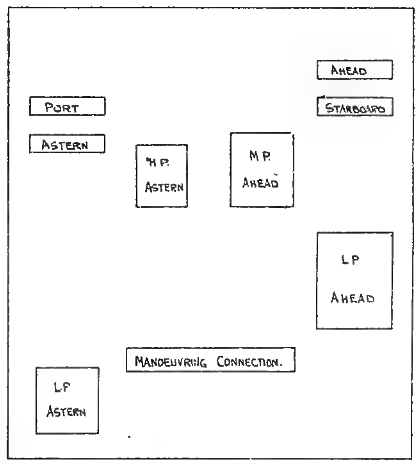
DIAGRAM OF ILLUMINATED MODEL BOARD.



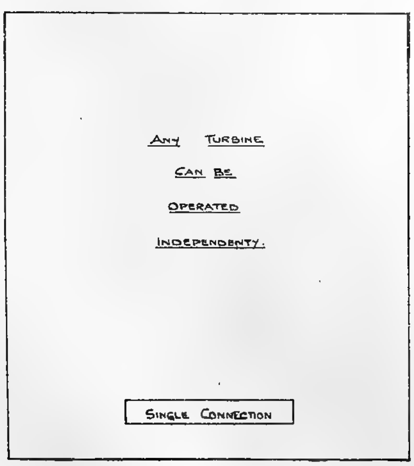
TRIPLE SERIES CONNECTION



MANOEUVRING:– (1) PORT - AHEAD. STARBOARD - ASTERN.



MANOEUVRING:– (2) PORT - ASTERN STARBOARD - AHEAD



SINGLE CONNECTION.

FIG. 12.

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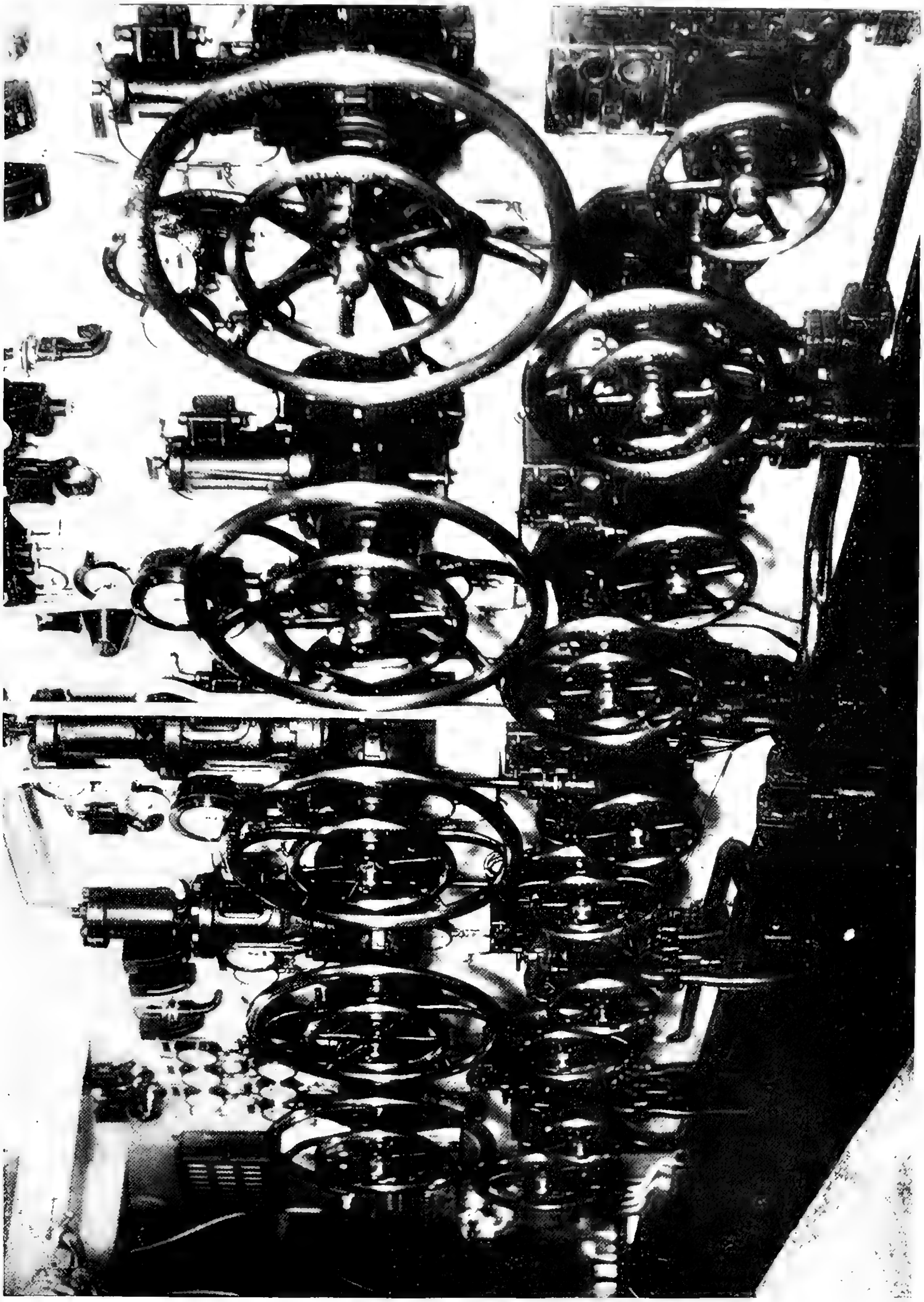
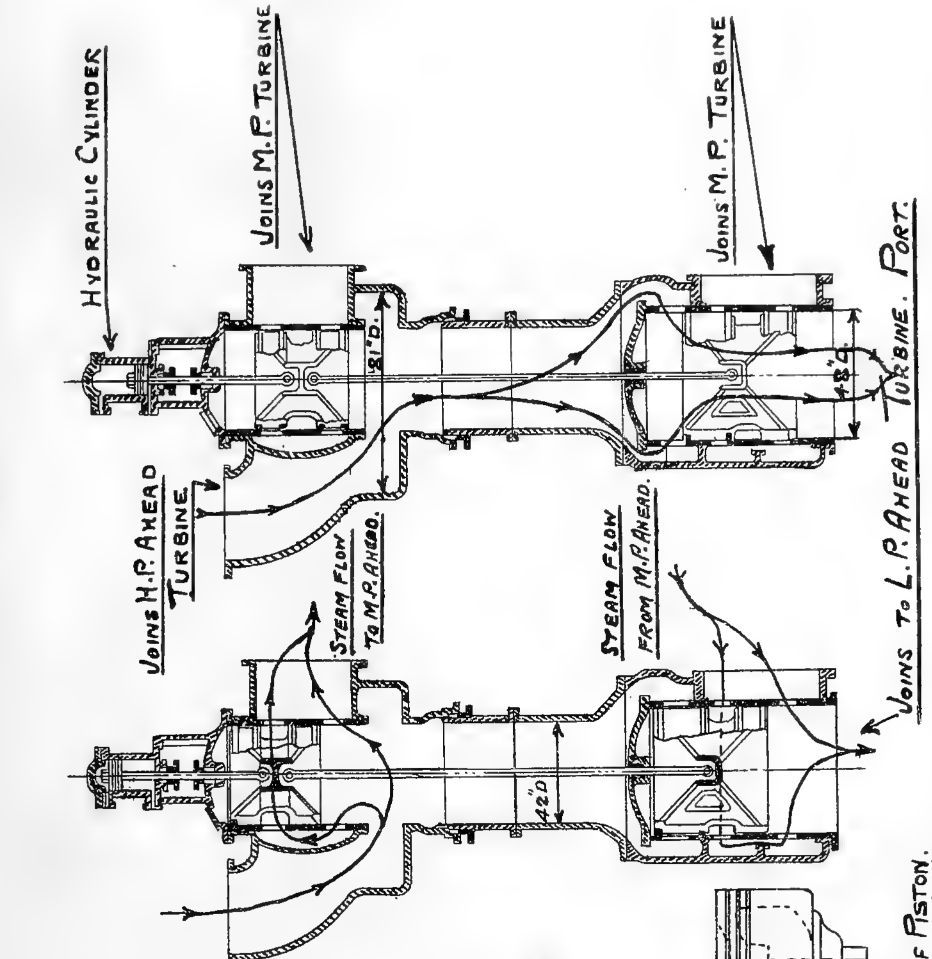


Fig. 13.

To illustrate paper on "The Propelling Machinery of the U. S. S. Leviathan," by Ernest H. B. Anderson, Member.

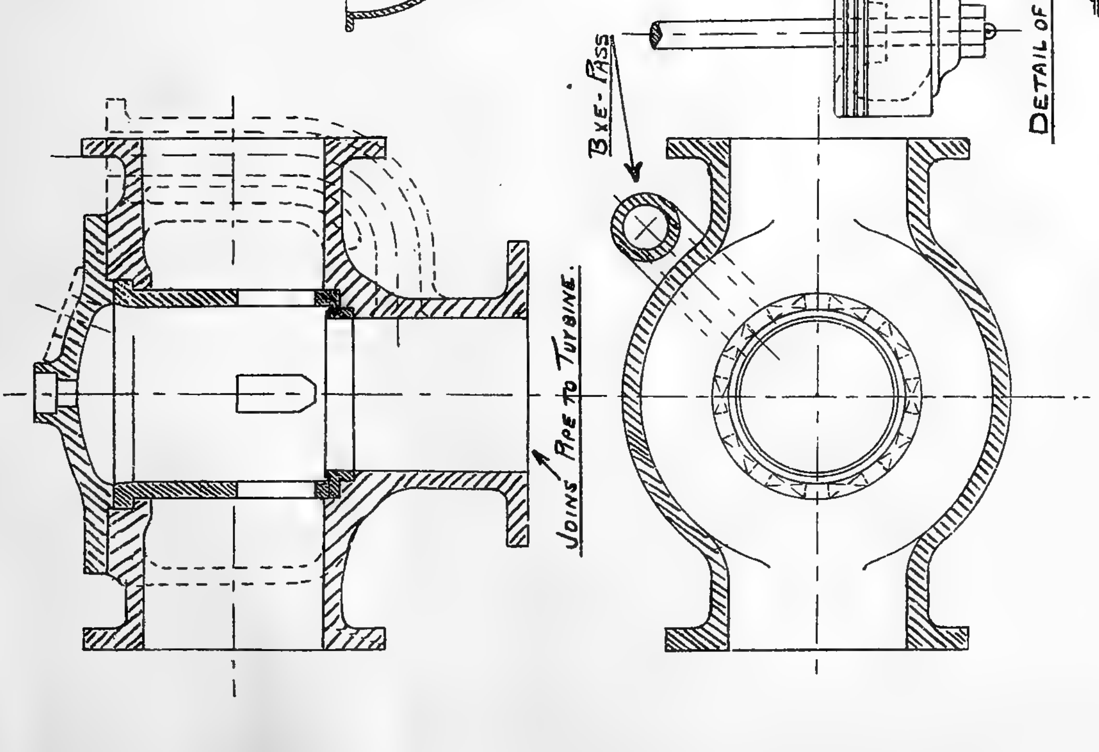
Mid Pressure Turbine Change Valve.



In Triple Series Connection. In Manoeuvring Connection.

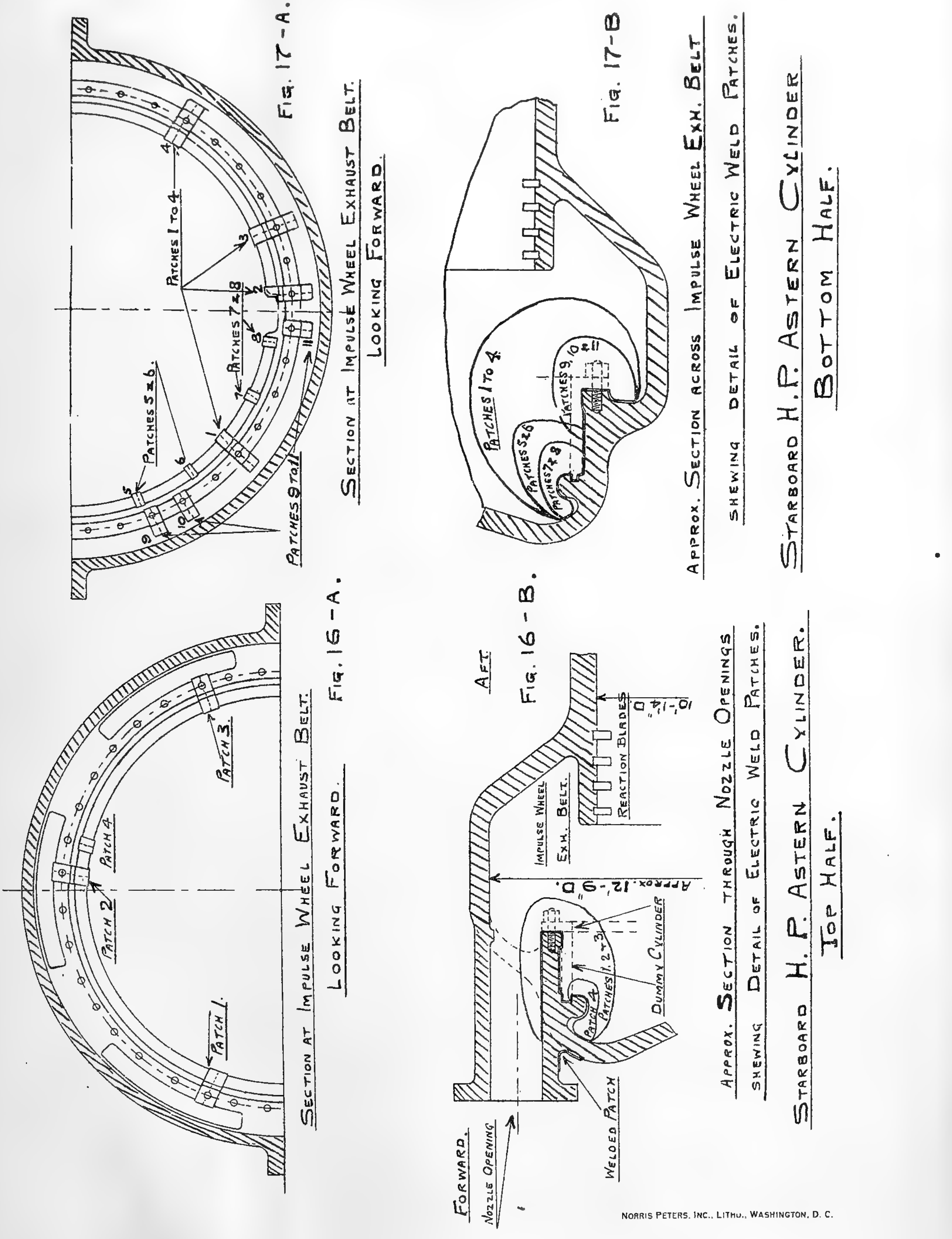
Fig. 15

Norris Peters, Inc., Litho., Washington, D. C.



Sketch of Turbine Regulator Valve. Fig. 14.

To illustrate paper on "The Propelling Machinery of the U. S. S. Leviathan," by Ernest H. B. Anderson, Member.



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To illustrate paper on "The Propelling Machinery of the U. S. S. Leviathan," by Ernest H. B. Anderson, Member.

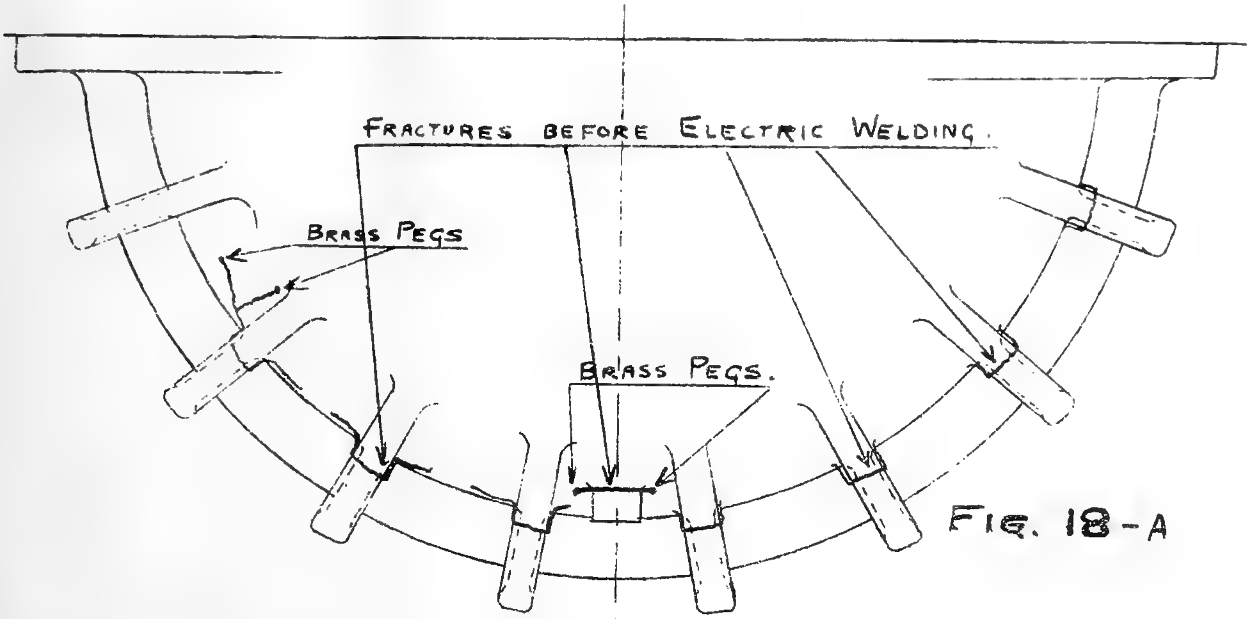

Fig. 18-A

External End Elevation of Casing

Looking Aft.

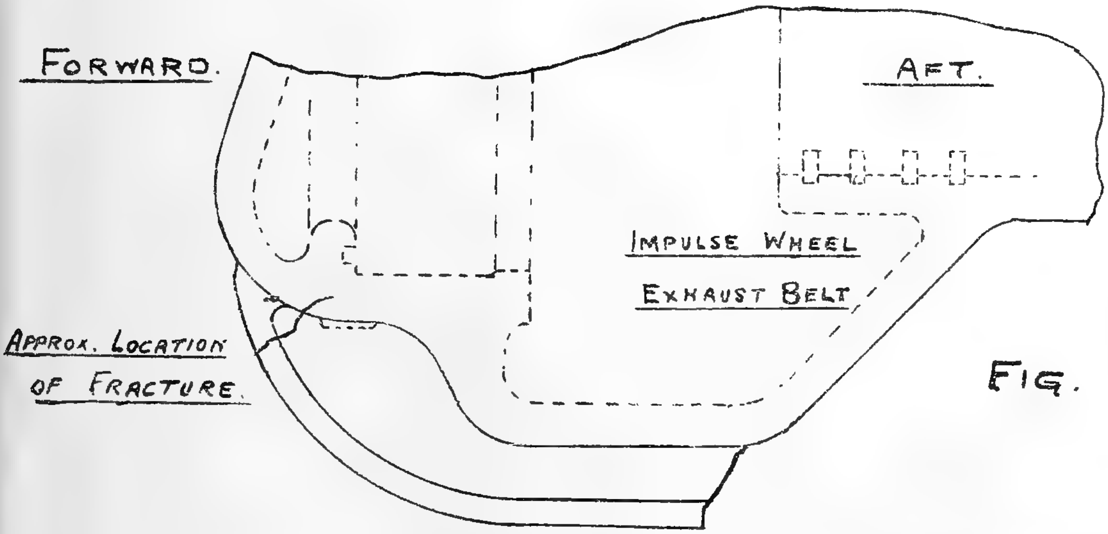

Fig. 18-B.

Approx. Side Elevation shewing Bulb Webs.

Starboard H.P. Astern Cylinder.

Lower Half.

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To ... ate paper on "The Propelling Machinery of the U. S. S. Leviathan."

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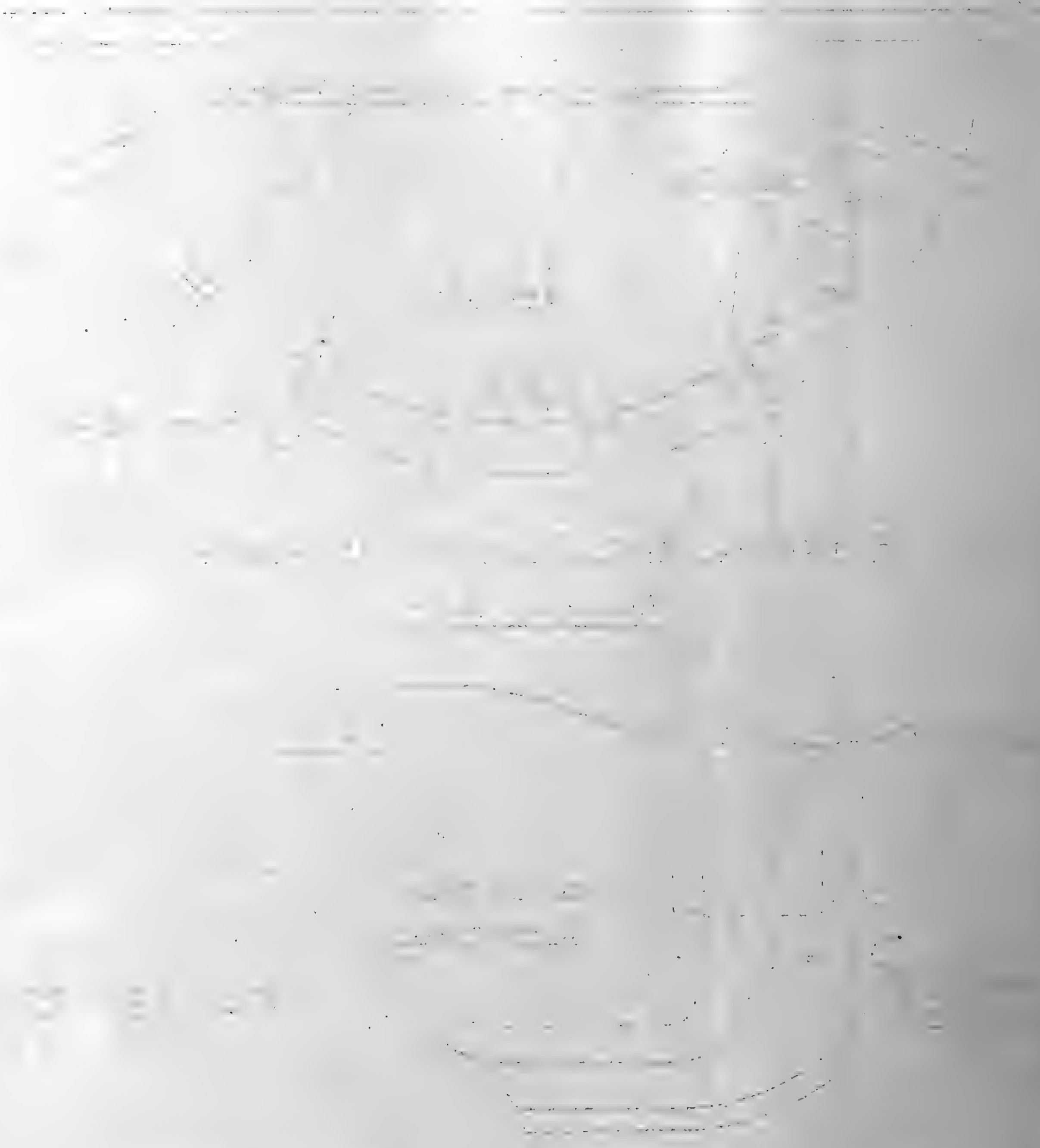
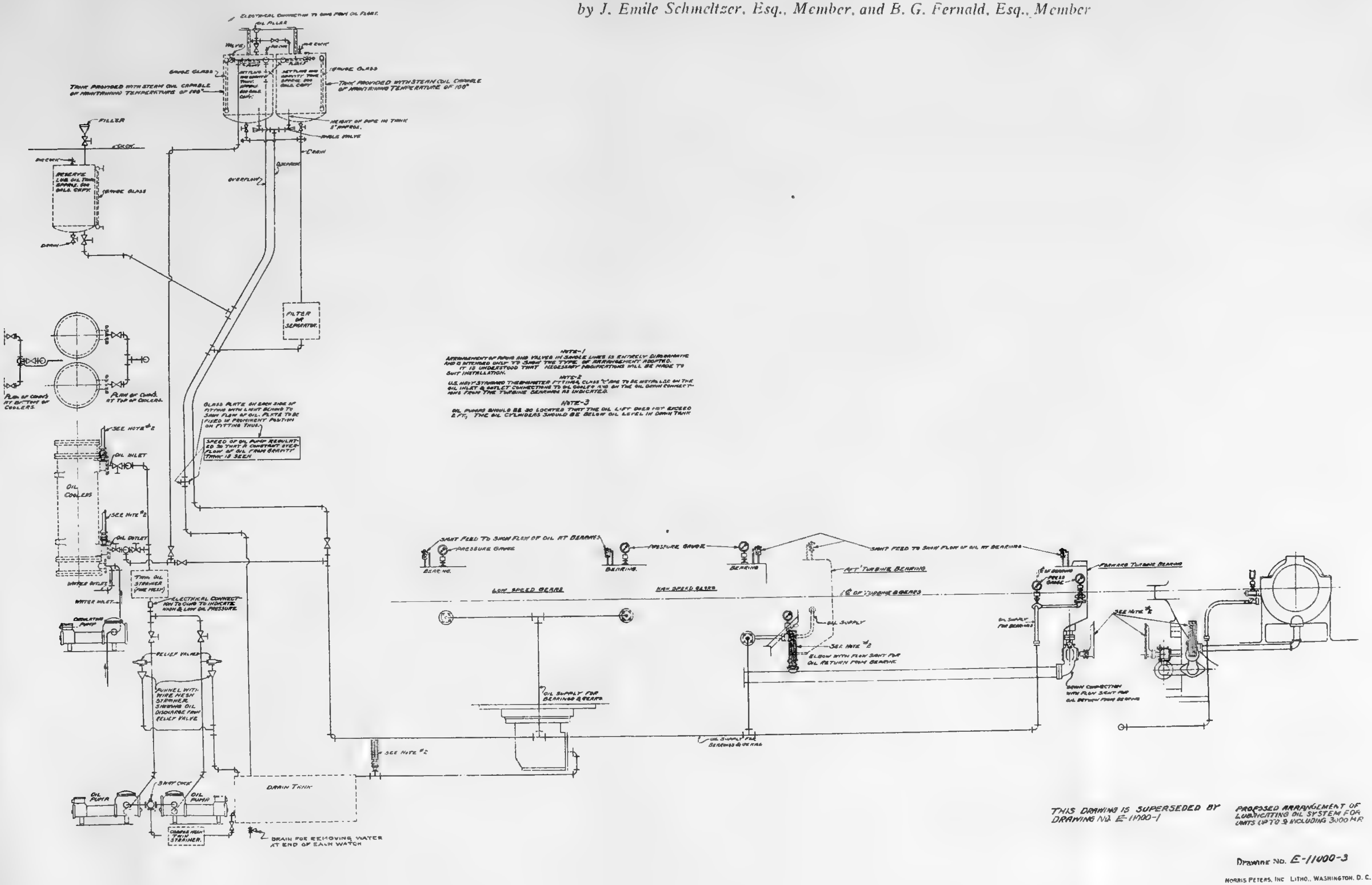
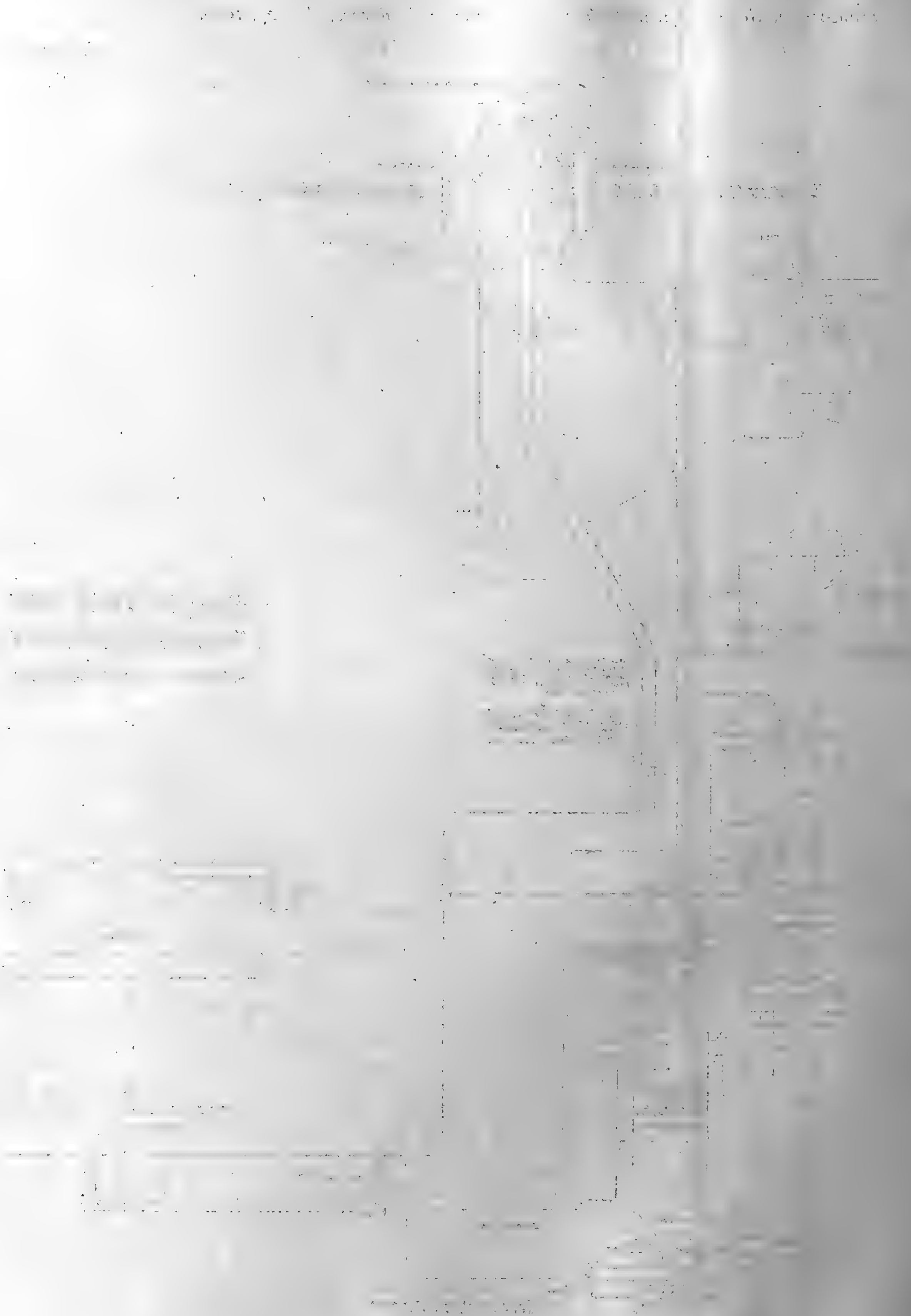


[illegible]

Transactions Society Naval Architects and Marine Engineers, Vol. 27, 1919.

To illustrate paper on "Standard Lubrication Oil System for Geared Turbines,"
by J. Emile Schmeltzer, Esq., Member, and B. G. Fernald, Esq., Member





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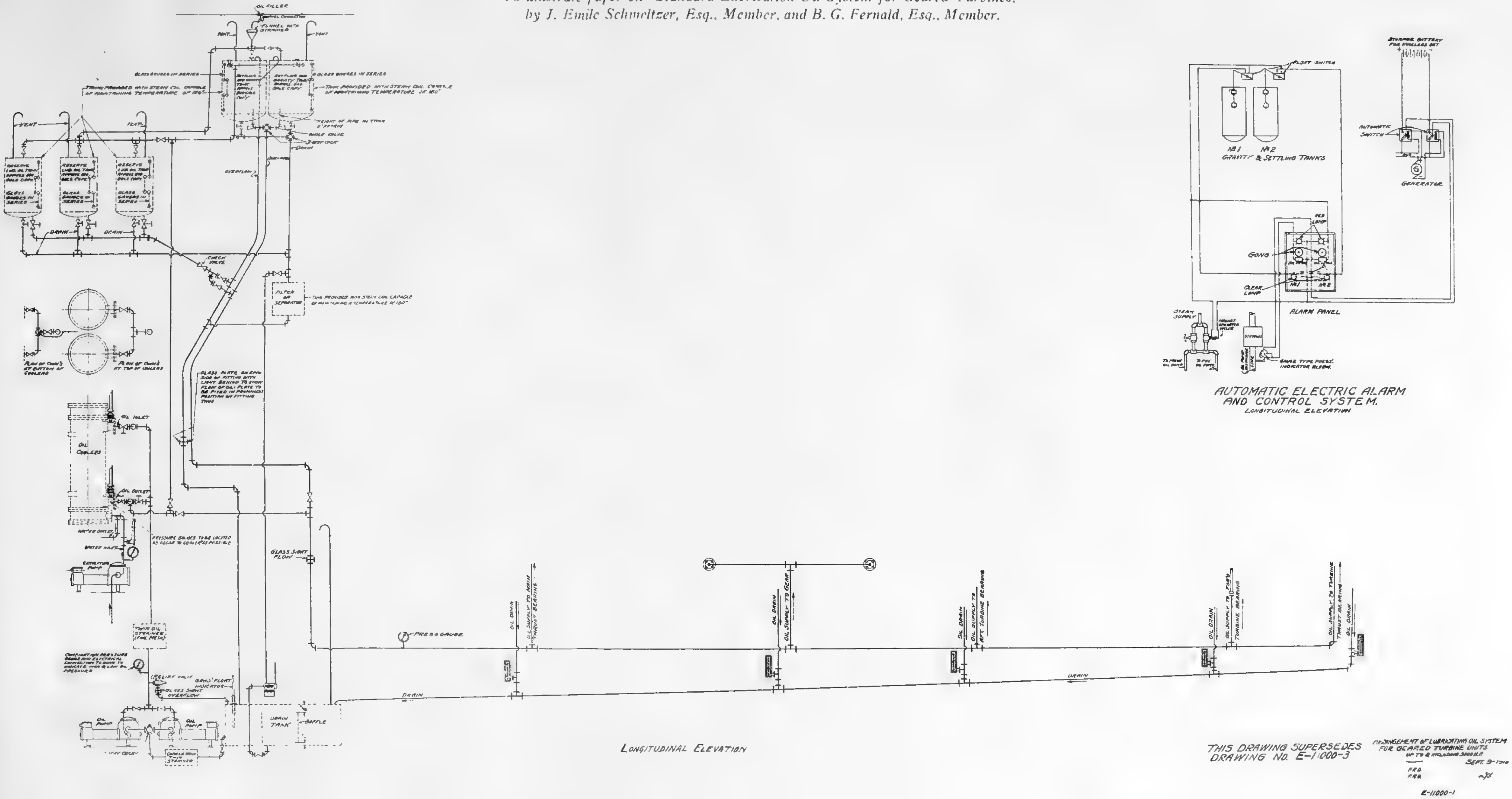

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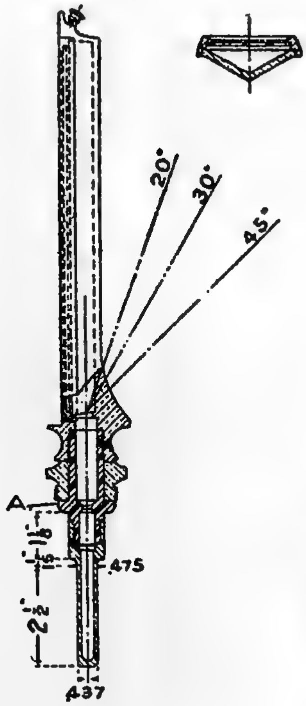
To illustrate paper on "Standard Lubrication Oil System for Geared Turbines," by J. Emile Schmeltzer, Esq., Member, and B. G. Fernald, Esq., Member.

Enclosure (c)

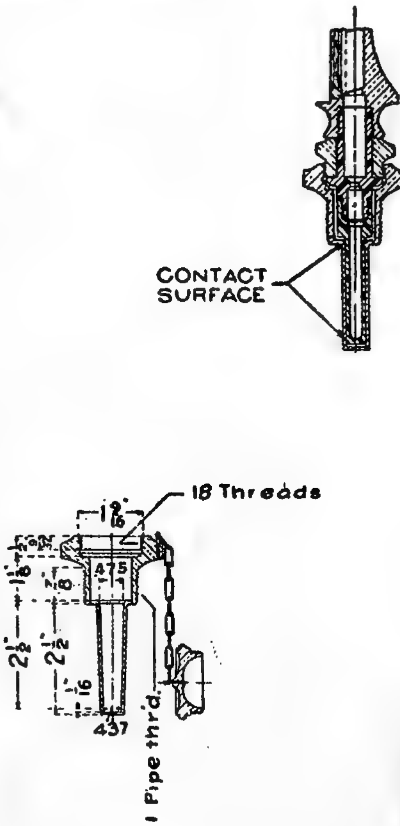
BUREAU OF STEAM ENGINEERING STANDARDS.

CLASS A.—THERMOMETER FITTINGS.

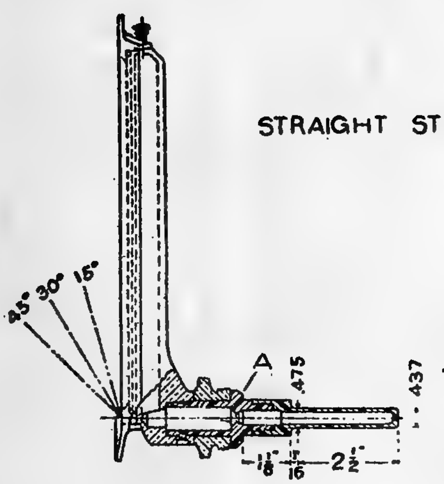
NOTE:- CONNECTING PIECE "A" TO BE COPPER PLATED, OR MADE OF COMPOSITION.



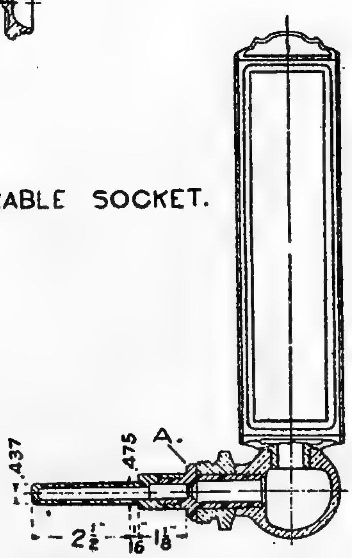
STRAIGHT STEM



4-INCH SEPARABLE SOCKET.



REGULAR 90° ANGLE STEM.



30° SIDE ANGLE STEM.

B–56

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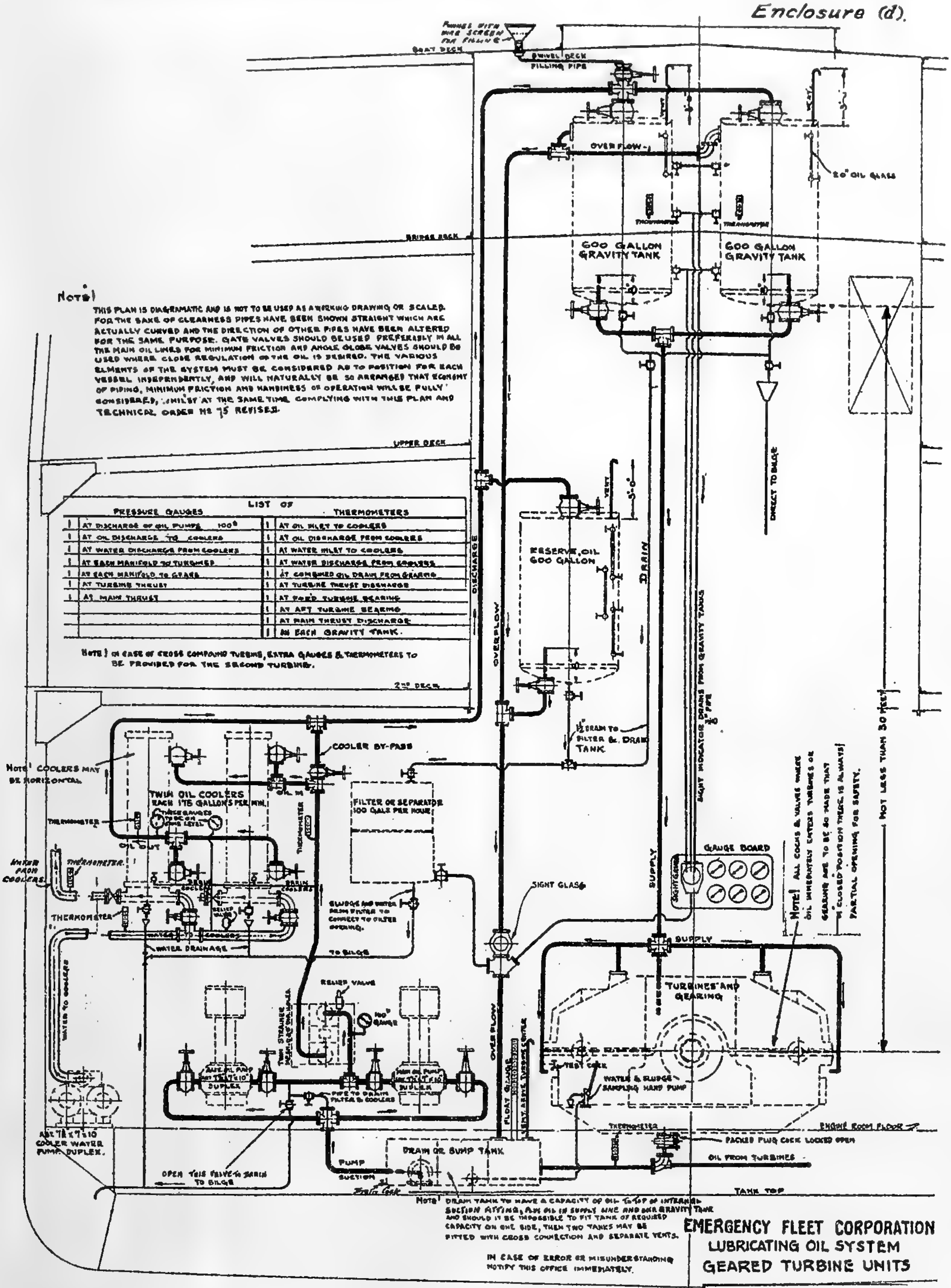
To illustrate paper on "Standard Lubrication Oil System for Geared Turbines," by J. Emile Schmeltzer, Esq., Member, and B. G. Fernald, Esq., Member

Enclosure (d).



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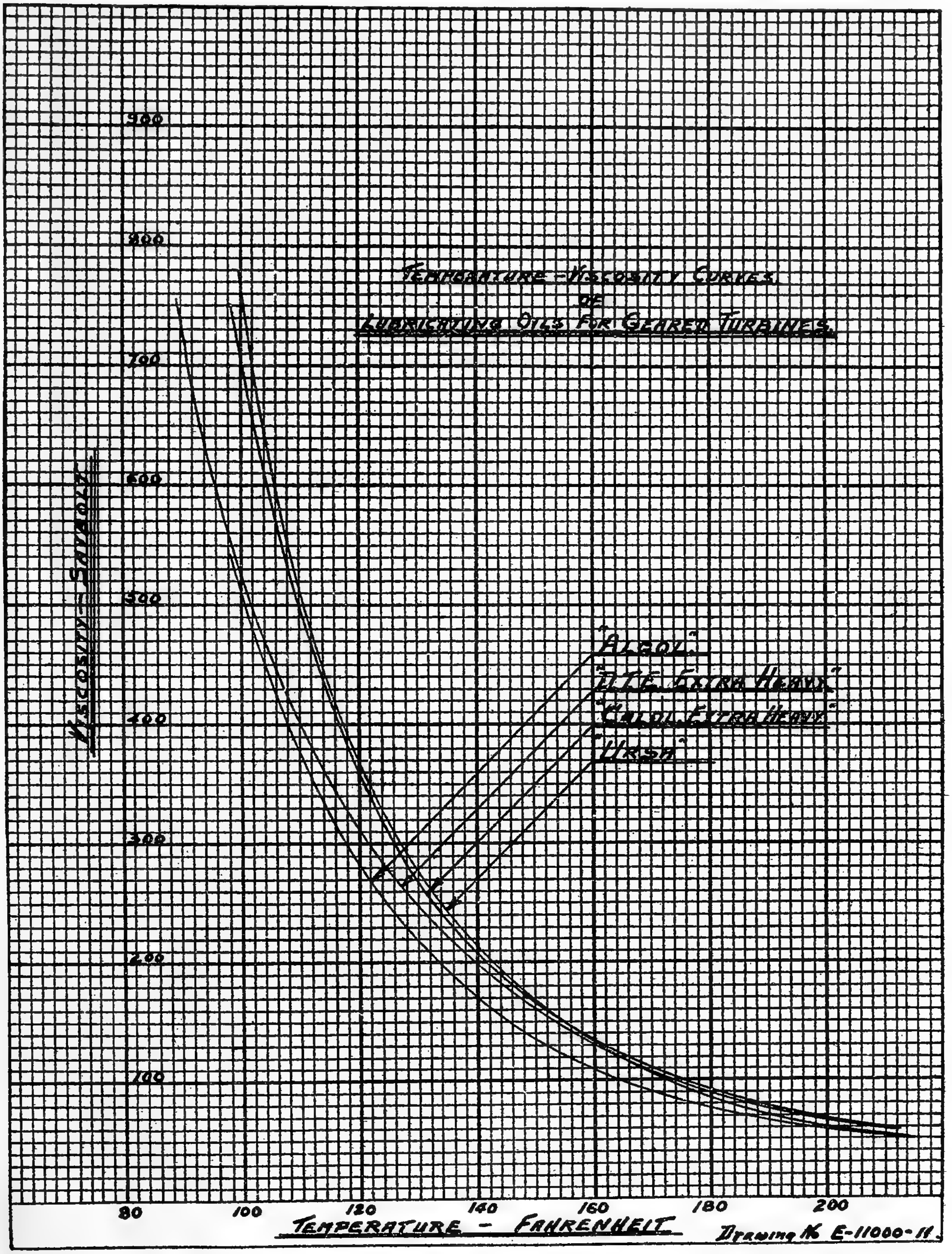

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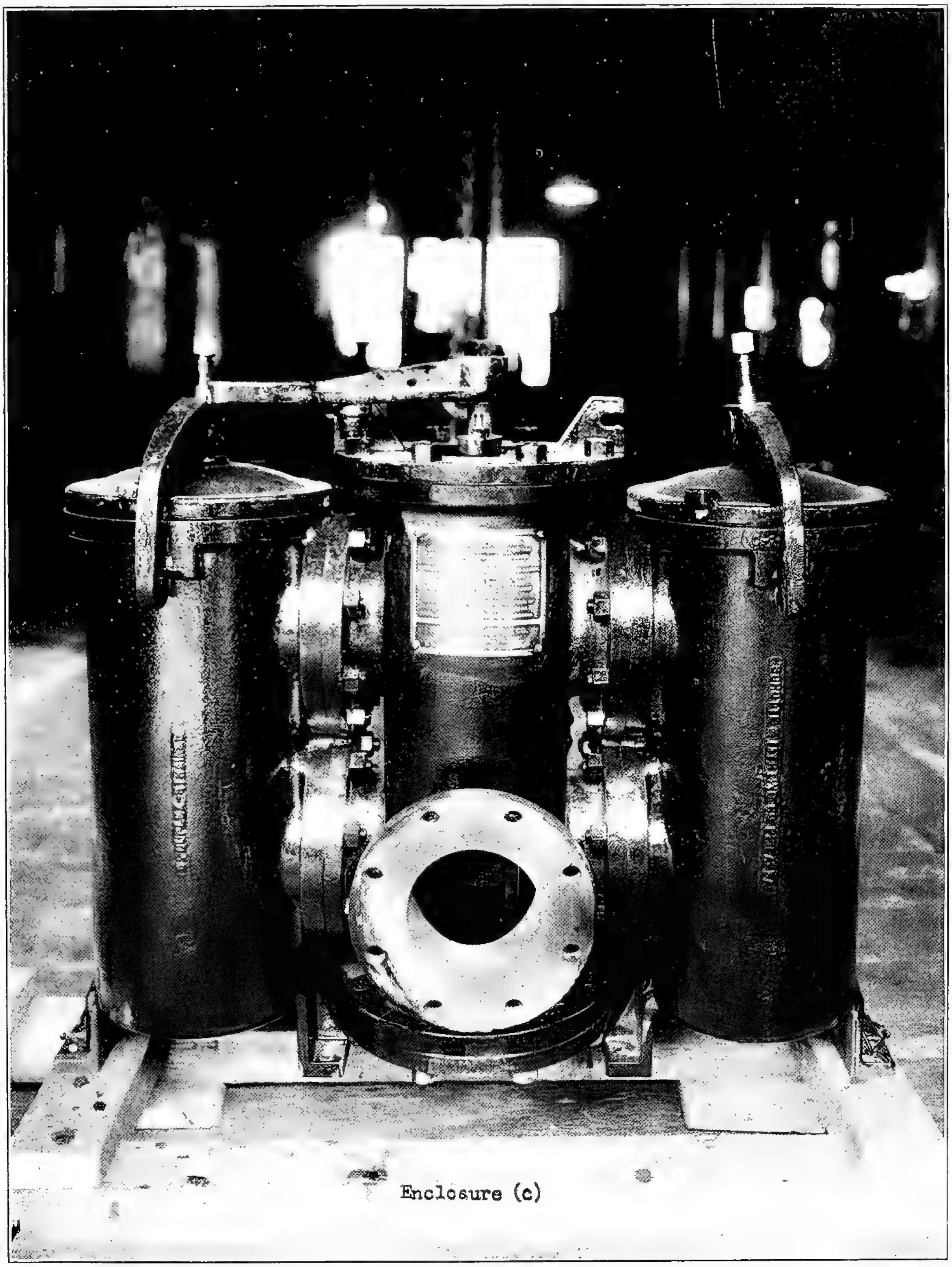
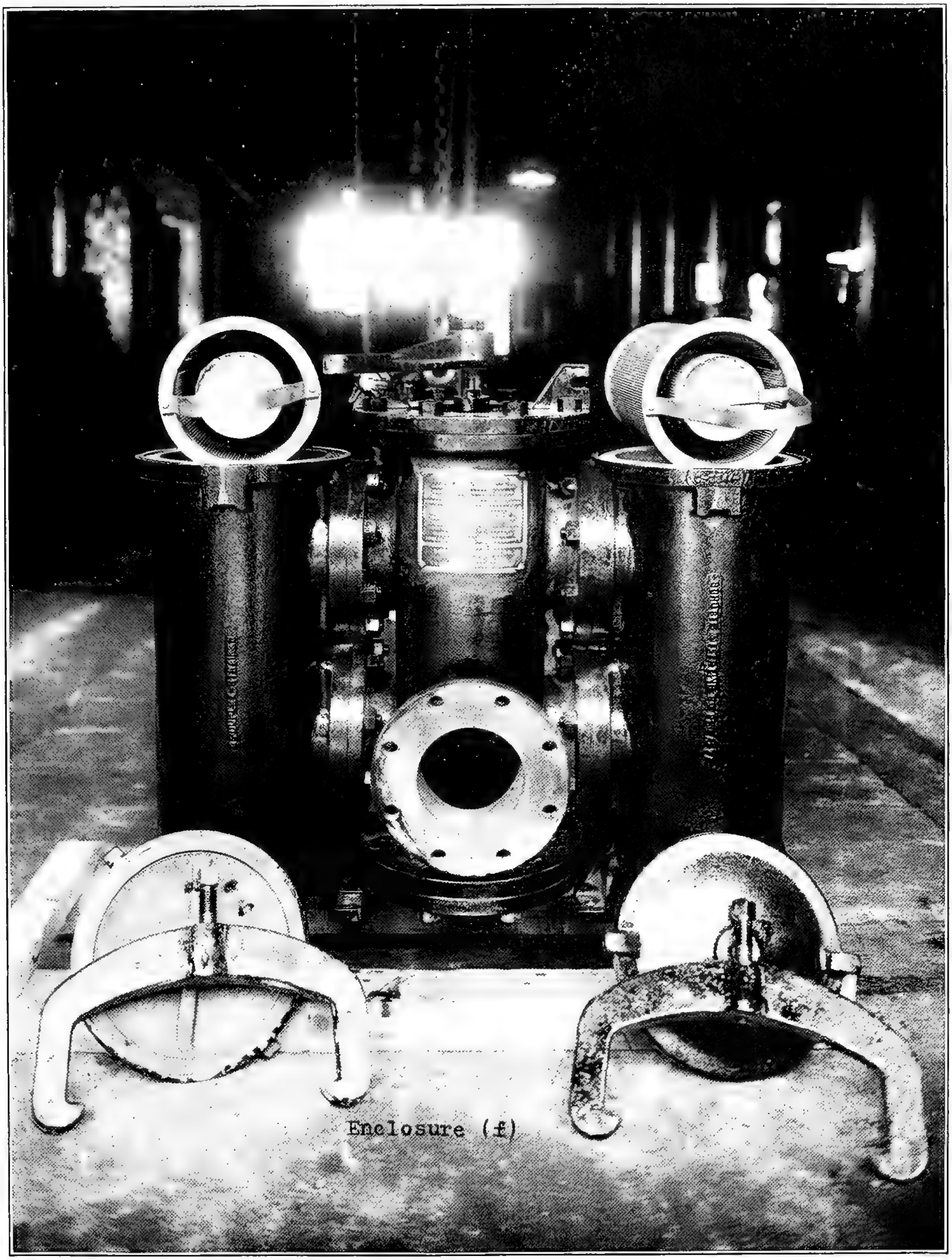


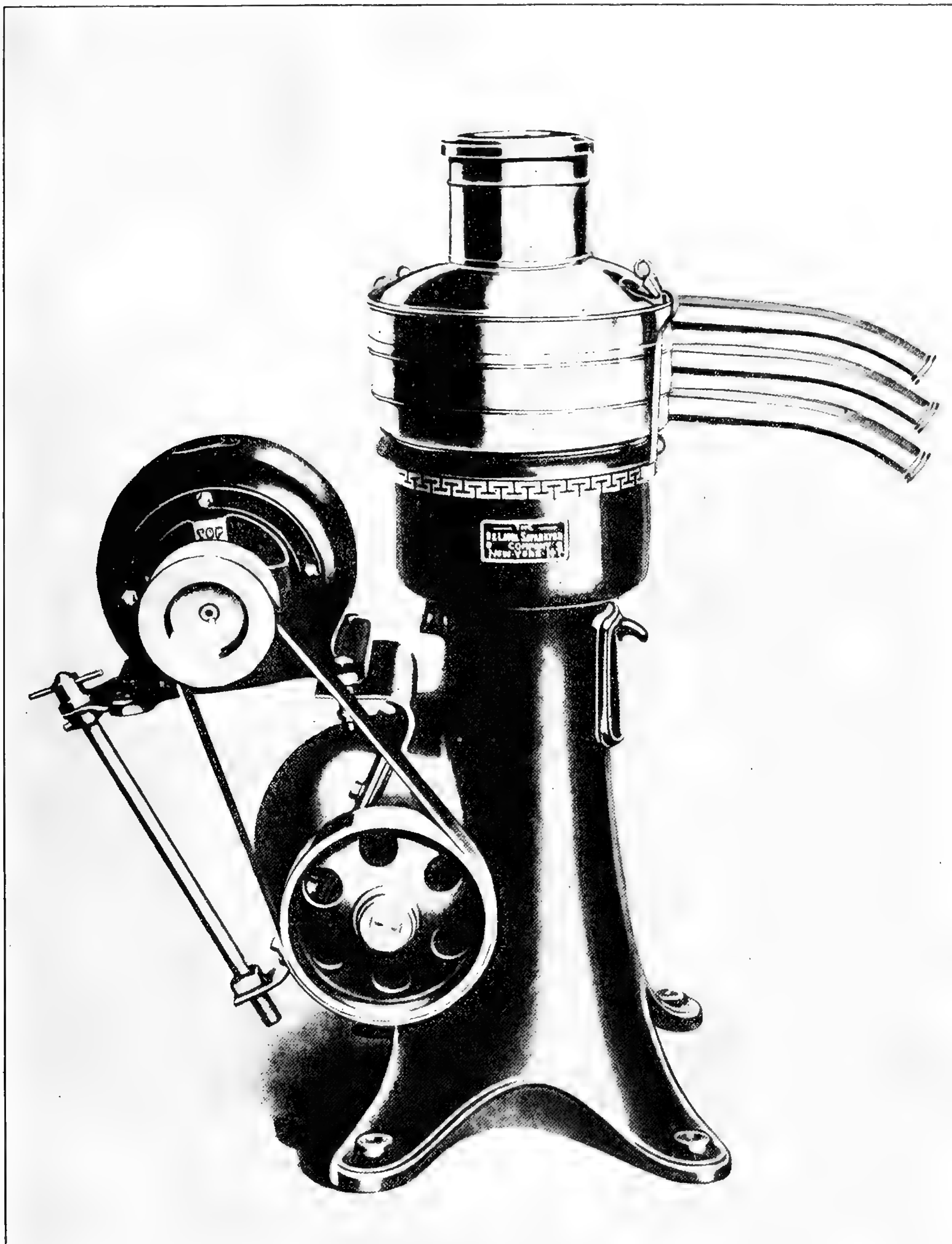
Discharge Strainer.

To illustrate paper on "Standard Lubrication Oil System for Geared Turbines," by J. Emile Schmeltzer, Esq., Member, and B. G. Fernald, Esq., Member.



DISCHARGE STRAINER DISASSEMBLED.

To illustrate paper on "Standard Lubrication Oil System for Geared Turbines," by J. Emile Schmeltzer, Esq., Member, and B. G. Fernald, Esq., Member.



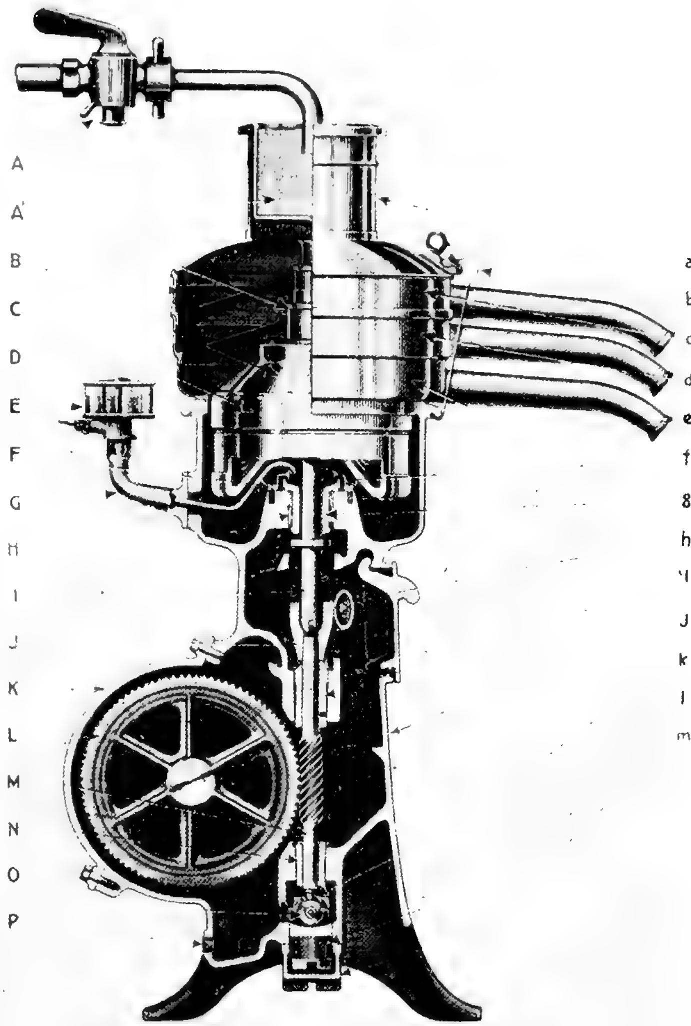
Enclosure (g)

CENTRIFUGAL SEPARATOR.

To illustrate paper on "Standard Lubrication Oil System for Geared Turbines," by J. Emile Schmeltzer, Esq., Member, and B. G. Fernald, Esq., Member.

NAMES AND NUMBERS OF PA

Letter	Name
A	Faucet
A1	Strainer
B	Top Disc
C	Bowl Top
D	Coupling R[illegible]
E	Lubricator (complete)
F	Screw for Top Bearing Dust Cap
G	Lubricator Arm (complete)
H	Top Bearing Spring
I	Bowl Sp[illegible]
J	Spindle Head
K	Worm Wheel Cap
L	Worm Wheel
M	Worm Screw Spindle
N	Lower Bushing
O	Tread Wheels (pair) [illegible]
P	Plug Screw for [illegible]
a	Regulat[illegible] Co[illegible]
b	Cover Clamp
c	Overflow Cover
d	Liquid Cover
e	Sludge Cover
f	Top Bearing Dust Cap
g	Top Bearing
h	Speed Indicator Wheel
i	Upper Bushing
j	Frame
k	Steel Point
l	Bottom Screw
m	Bottom Screw Lock Nut



Be careful to distinguish between C

SECTIONAL CUT OF THE DE LAVAL "NO. 300" SPECIAL CENTRIFUGAL SEPARATOR.

THE DE LAVAL SEPARATOI

THE DE LAVAL SEPARATOR CO., NEW YORK.

Enclosure (h)

CENTRIFUGAL SEPARATOR.

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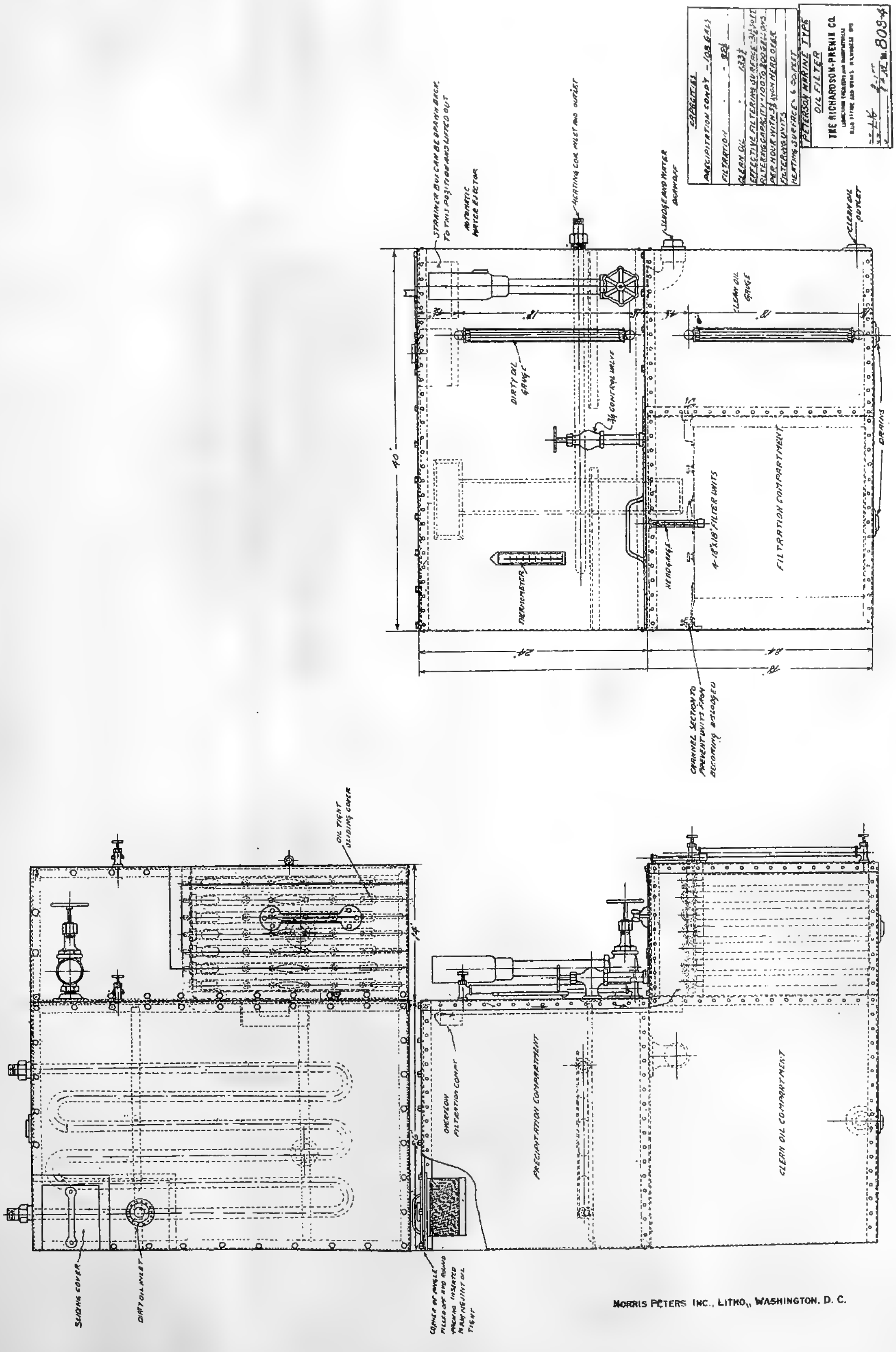

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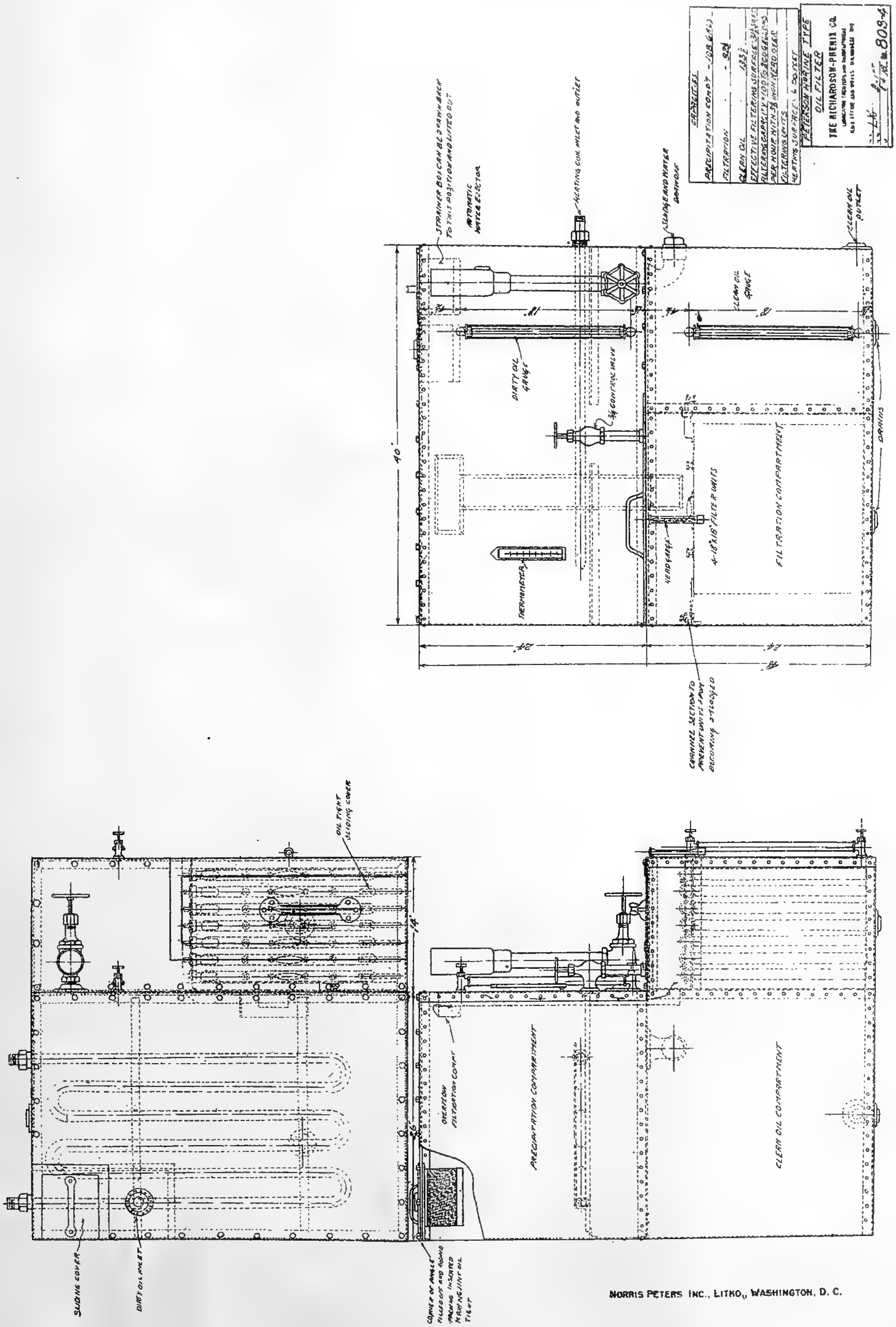

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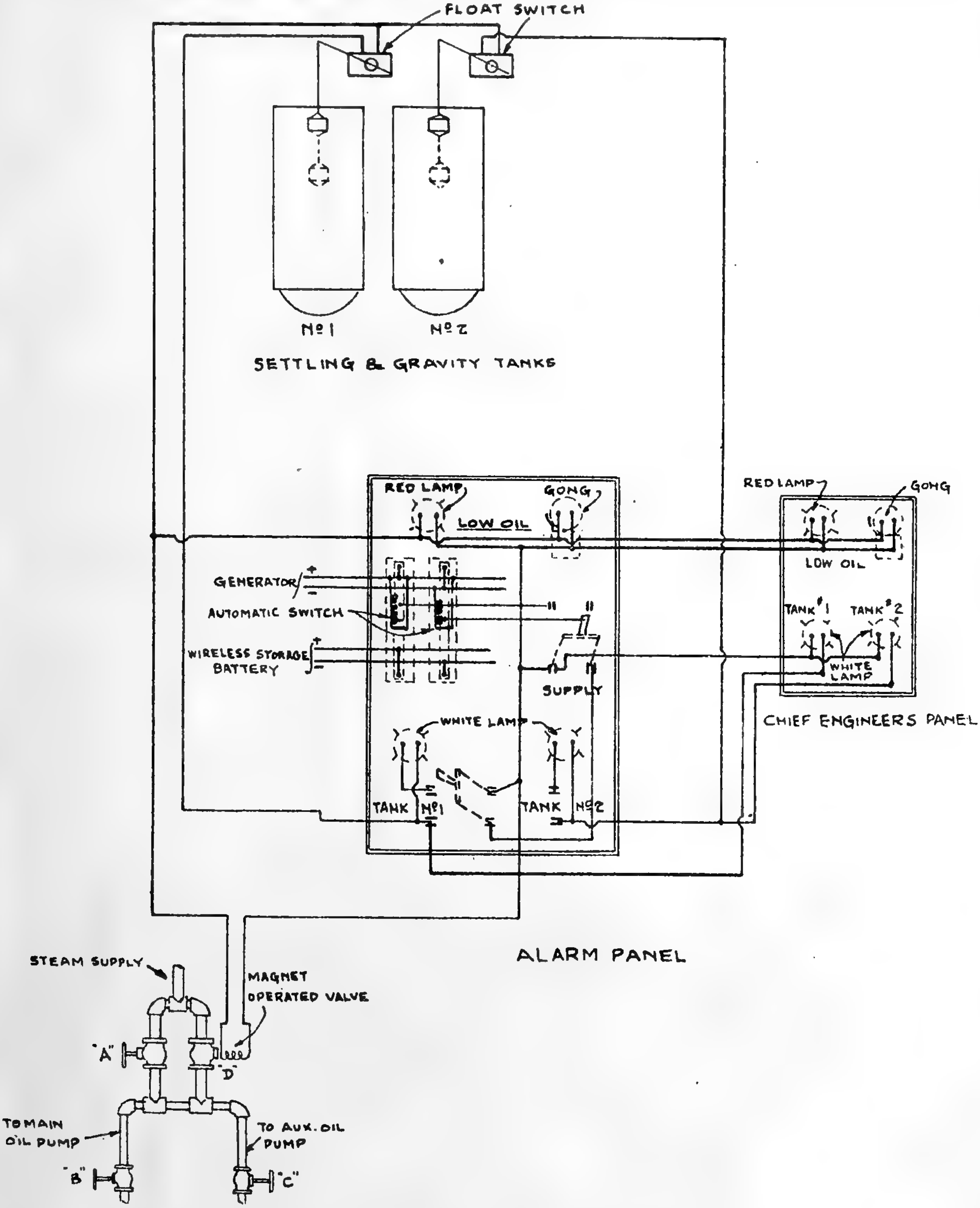
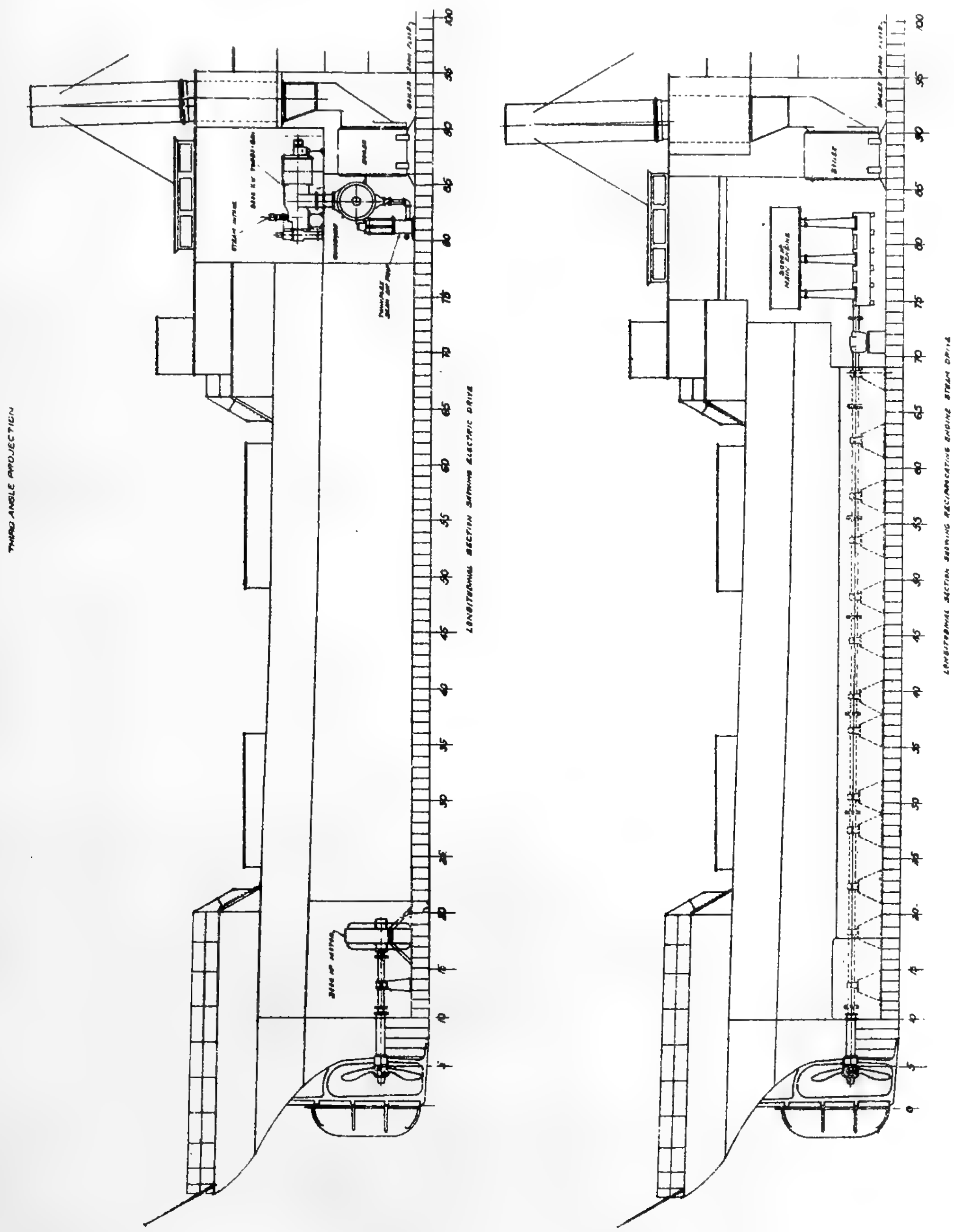


AUTOMATIC ELECTRIC ALARM CONTROL SYSTEM

DRAWING No E-11000-11.

NORRIS PETERS, INC., LITHO., WASHINGTON, D. C.

To illustrate paper on "Electric Propulsion of Merchant Ships," by W. L. R. Emmet, Esq., Member of Council.



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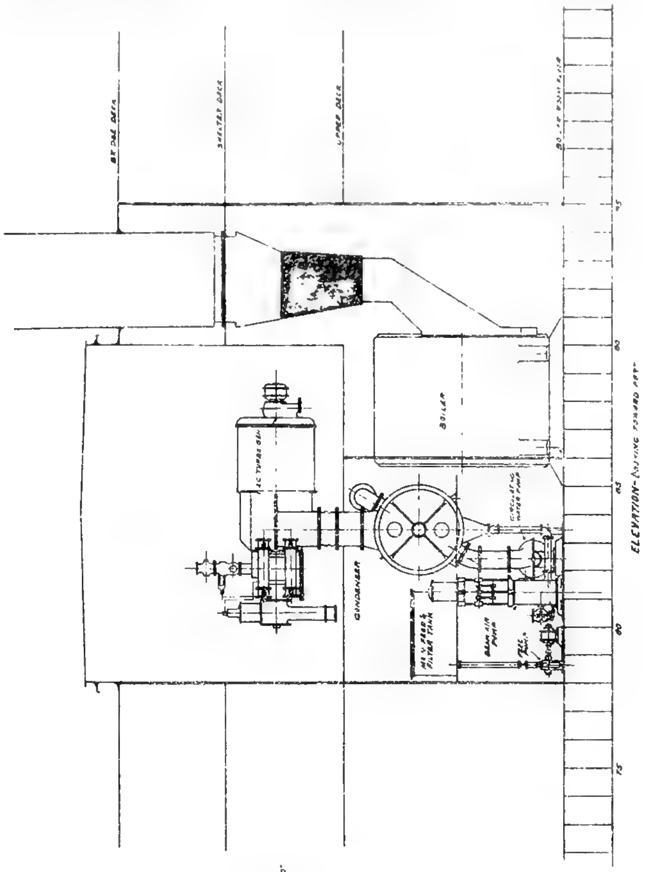

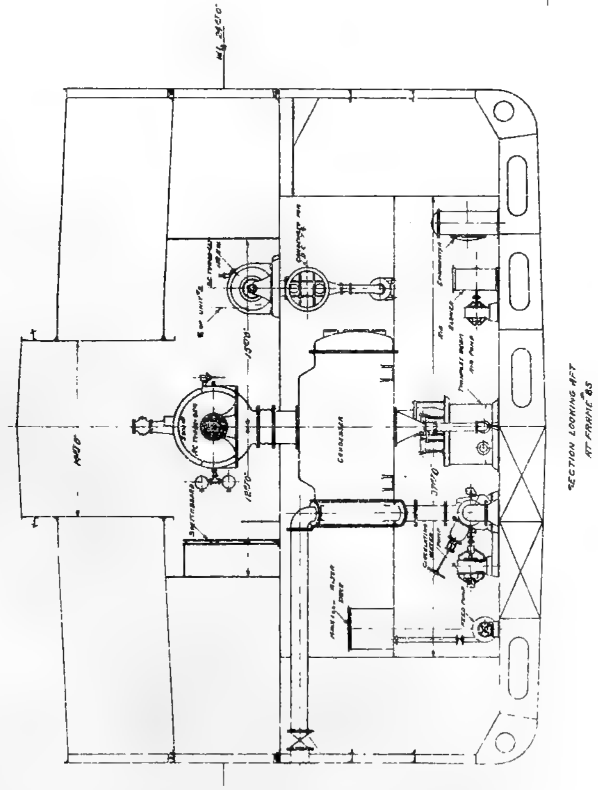

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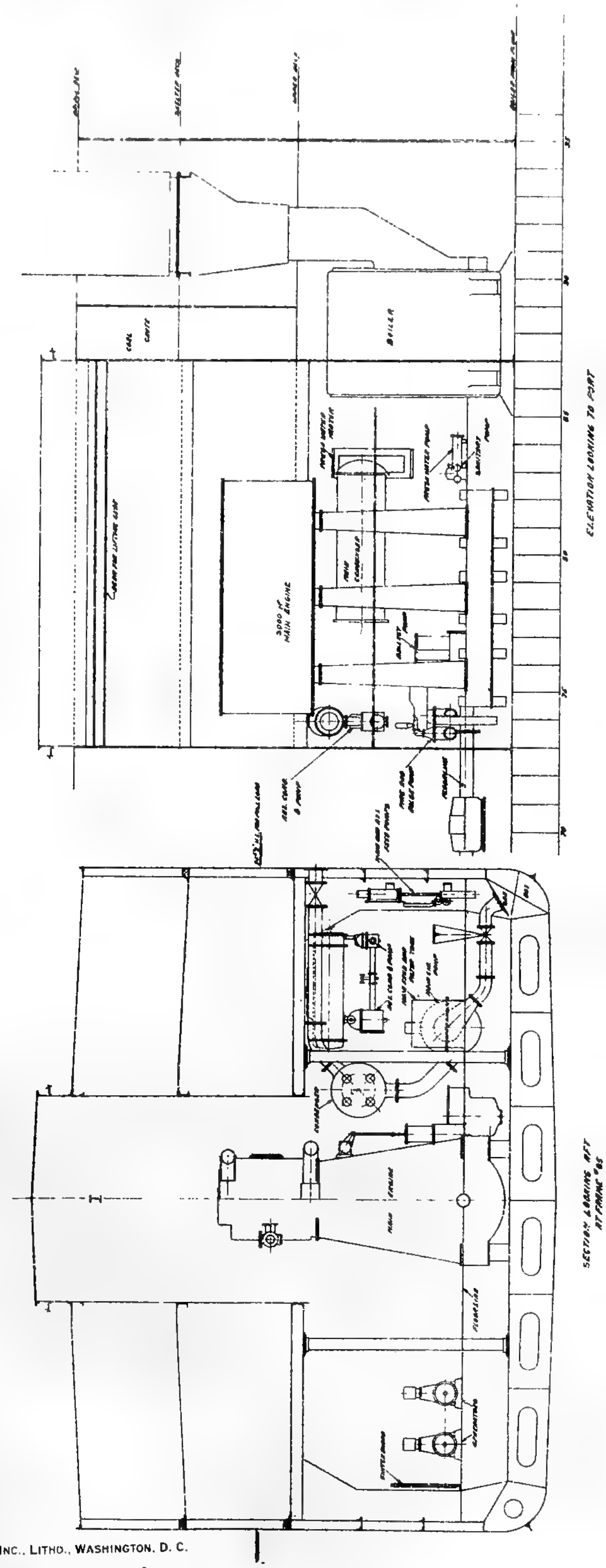

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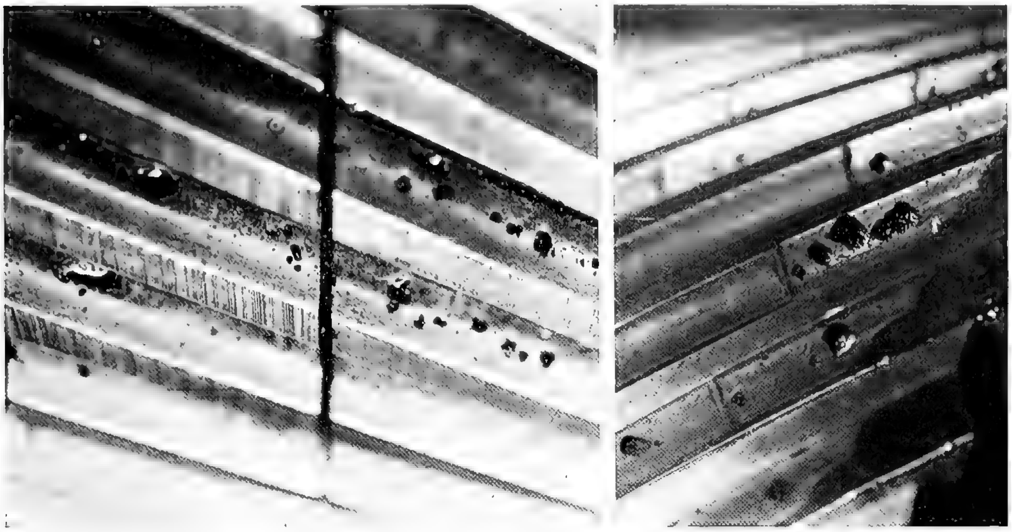
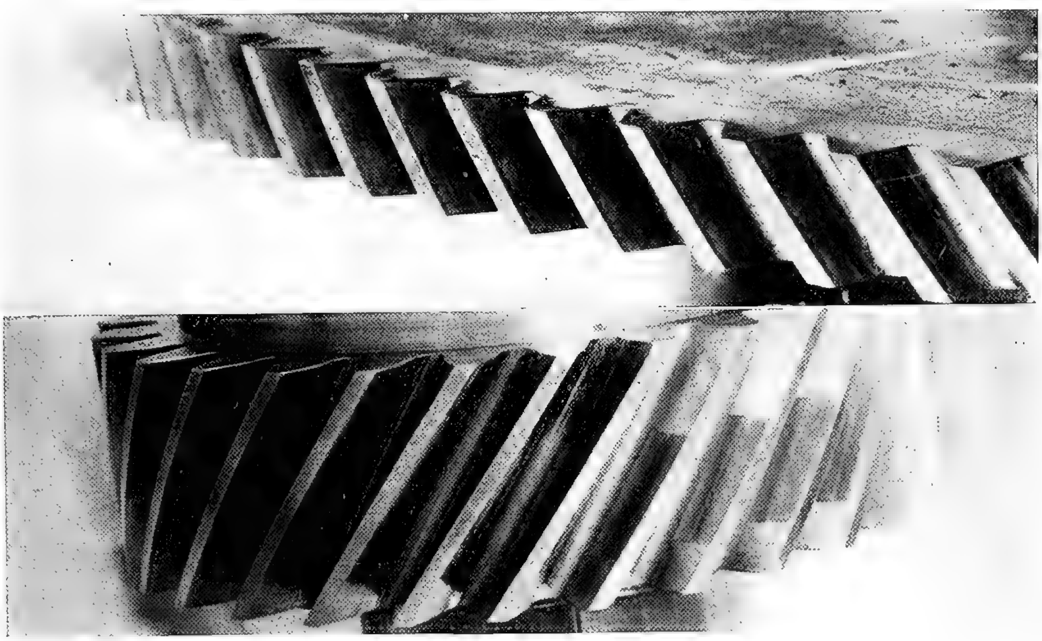
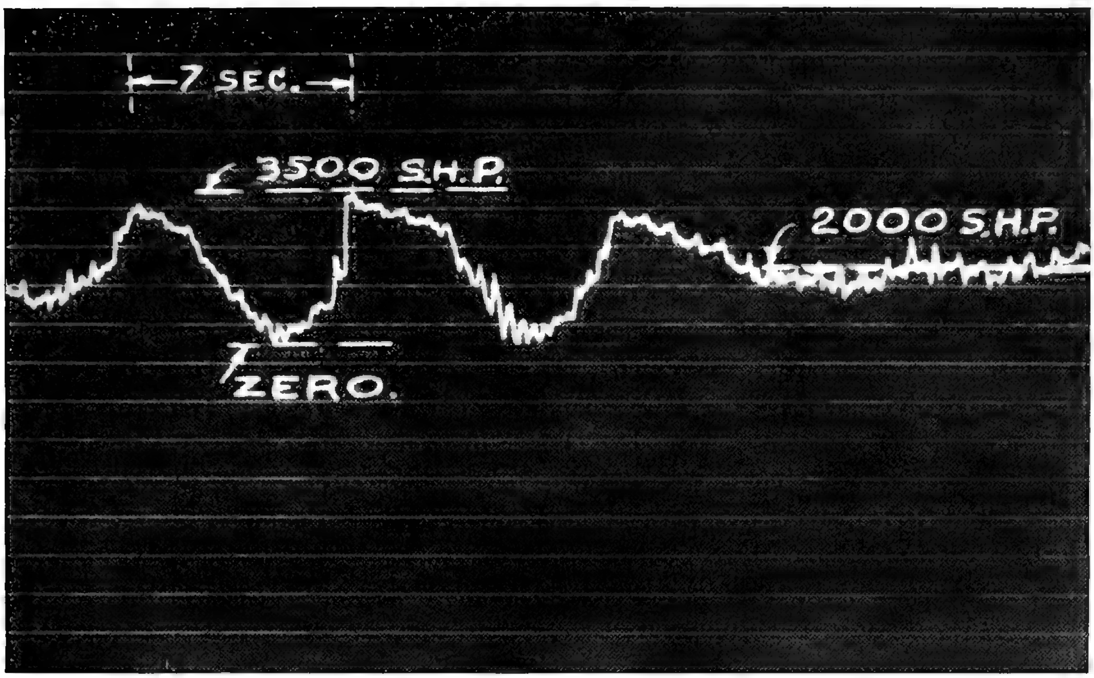


Low Speed Ship Gear and Pinion. Pinion Diameter 11.44 Inches. Pitch 3½ Inches. Normal Load 1,150 Pounds Per Inch Face. Tooth Speed 1,272 Feet Per Minute. Time Run, About 400 Hours at Sea.



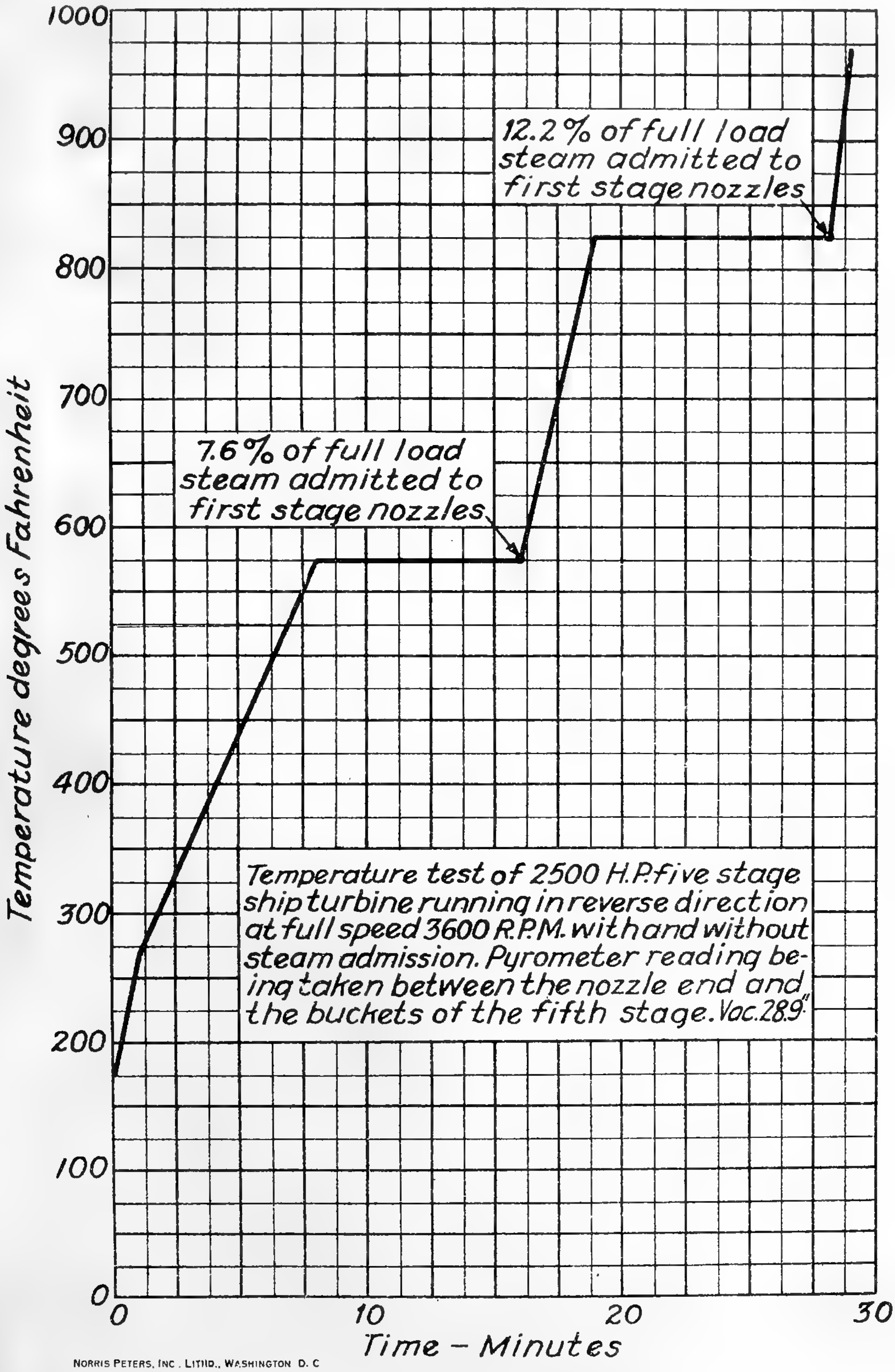
Experimental Gear Disc and Pinion. Pinion Diameter 7.28 Inches. Pitch 4 Inches. Load Carried 3,000 Pounds Per Inch Face. Tooth Speed 7,000 Feet Per Minute. Time Run 263 Hours in Schenectady. Has Made 8 Times as Many Tooth Engagements as Above With a Pressure, Which, Considering Smaller Pinion Diameter, is Relatively 4 Times as Heavy.

To illustrate paper on "Electric Propulsion of Merchant Ships," by W. L. R. Emmet, Esq., Member of Council.



Record From Torsion Spring Coupling on S. S. Jebsen in Ballast in a Moderately Rough Sea. Average R. P. M. 78. Average Shaft Horse-power About 2,000. Part of the Smaller Fluctuations Shown Came From an Untrue Collar in the Instrument, Otherwise Record is Correct.

To illustrate paper on "Electric Propulsion of Merchant Ships," by W. L. R. Emmet, Esq., Member of Council.



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To illustrate discussion by Francis Hodgkinson, Esq., on paper entitled "Electric Propulsion of Merchant Ships," by W. L. R. Emmet, Esq., Member of Council.